FABRICATION OF COMPRESSOR TEST RIG

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JANUARY, 1973

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A Thesis Submitted
In Partial Fulfilment of the Requirements
for the Degree of
MASTER OF TECHNOLOGY

By
KAMESHWAR PRASAD

to the

DEPARTMENT OF MECHANICAL ENGINEERING
INDIAN INSTITUTE OF TECHNOLOGY KANPUR
JANUARY, 1973

To

My parents, who have suffered and sacrificed for my education.

My wife, who has taken a good deal of pain for sparing time for the present work.

My sons, who have suffered their education during this period.

Kameshwar Prasad

CERTIFICATE

This is to certify that this work on "Fabrication of Compressor Test Rig" has been carried out under my supervision and has not been submitted elsewhere for a degree.

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ACKNOWLEDGEMENT

The author expresses his extreme sense of gratitude and indebtedness to Professor V. Kadambi for his valuable guidance and constant encouragement throughout this work.

The author is very grateful to the Bihar Government for deputing him to I.I.T. Kanpur for higher studies.

The author is grateful to Mr. M. Prasad and Mr. N.K. Giri for useful discussions and various suggestions during different stages of this work and to Mr. Mithileshwar Prasad for useful discussions whenever required.

The author also wishes to put on record his gratefulness to

Mr. S.K. Chaturvedi and N. Ravichandran for their help in taking the readings and discussions at different stages,

Mr. P.N. Mishra for his very sincere help and initiative in the fabrication and setting up the experimental set-up,

Mr. Mushtaq Ali and Mr. D.K. Sarkar for their help when required at different stages of the work,

and to his friends for their timely help without which the work would have been incomplete.

Last but not the least, the author would like to express his appreciation and gratitude to Mr. J.D. Varma for his excellent typing.

CONTENTS

LIST OF FIGURES

NOMENCLATURE

$\dot{m}_f$	=	Mass flow rate of the refrigerant (kg/min.)
ϕ_h	=	Rate of energy input to the calorimeter (kcal/min.)
ϕ_s	=	The isentropic work of the compressor in compressing the saturated vapour at the outlet of the evaporator to the condenser presser (kcal/kg)
ϕ_i	=	Heat input to the calorimeter while calibrating (kcal/min.)
F_l	=	Heat leakage factor (kcal/min, °C)
t_a	=	Ambient temperature (°C)
t_r	=	Refrigerant temperature (°C)
h_{g2}	=	Enthalpy of saturated vapour at the evaporator pressure (kcal/kg.)
h_{g1}	=	Enthalpy of the refrigerant at evaporator outlet (kcal/kg.)
h_{f2}	=	Enthalpy of liquid refrigerant before throttling (kcal/kg)
h_{f1}	=	Enthalpy of saturated liquid at compressor discharge pressure.
COP	=	Actual co-efficient of performance.
$(COP)_s$	=	Co-efficient of performance of the simple vapour compression refrigeration cycle (suction dry saturated)
$(COP)_R$	=	Relative co-efficient of performance ((COP) / $(COP)_s$)

kW_1	=	Power input to the Calorimeter (kW)
kW_2	=	Power input to the motor (kW)
$SKWC_2$	=	Power input to the shaft of the compressor (kW)
KWC_2	=	Power output of the motor shaft (kW)
L_2	=	Losses in the motor (kW)
RF	=	Refrigeration capacity (kcal/min.)
$(RF)_s$	=	Specific Capacity of the Compressor (kcal/kW hr)
P_1	=	Compressor discharge pressure (PSI)
P_2	=	Condenser inlet pressure (PSI)
P_3	=	Condenser outlet pressure (PSI)
P_4	=	Pressure of the refrigerant before exp. valve (PSI)
P_5	=	Evaporator outlet pressure (PSI)
TC_1	=	Compressor discharge temperature (mV)
TC_2	=	Condenser inlet temperature (mV)
TC_3	=	Condenser outlet temperature (mV)
TC_4	=	Temperature before exp. Valve (mV)
TC_5	=	Temperature just after exp. valve (mV)
TC_6	=	Temperature at the suction of the compressor (mV)
RM	=	Water flow rate in the condenser (gpm)
PC	=	Calibrated pressures at different points (PSI)
TCC	=	Calibrated temperatures at different points (°C)
RMC	=	Calibrated water flow rate in the condenser (kg/min)

CHAPTER 1

1.1 INTRODUCTION

As is well known the compressor is the most essential of the five different parts of the vapour compression refrigeration system which consists of : (i) the compressor, (ii) the condenser (iii) the evaporator, (iv) the expansion valve and (v) the connecting pipes. The sole responsibility for producing the desired refrigeration effect in the cooler is that of compressor. The performance of the whole plant depends upon it, and a knowledge of compressor performance can be used to predict the plant performance with greater accuracy.

The widely accepted definition of testing of refrigerant compressor is "To find the performance of the plant in which the compressor has been used" [1]. The performance of a machine is defined as "An evaluation of the ability of the machine to accomplish the assigned task. It is a design compromise between physical limitations of (a) refrigerant, (b) compressor and (c) motor to provide (i) the most refrigerating effect for the least power input, (ii) the greatest trouble free life, (iii) lowest cost, (iv) wide range of operating conditions". [1]

Measures of compressor performance are : (1) capacity which is proportional to the displacement and (2) performance factor. Capacity is the refrigeration effect that can be produced by the compressor. It is the difference between the total enthalpies of the refrigerant at the temperature corresponding

to the pressure of the vapour leaving the compressor and of the refrigerant vapour entering the compressor. The performance factor of open - type compressor is given by the equation :

Performance factor* (open) = (Capacity) / (Power input to the shaft)

The performance factor for a hermetic compressor includes motor efficiency and hence the performance factor of a hermetic compressor is given as follows :

Performance factor (hermetic) = (Capacity) / (Power input to the motor)

There are two types of compressor tests. The first is the 'life testing'. It is to determine the probable life of the machine by conducting tests under conditions simulating those under which the compressor must operate. This test does not include the determination of refrigeration capacity and the power input. It takes not less than a year to conduct this test. The second method of testing is the determination of capacity and co-efficient of performance. This method is also known as 'run round the cycle test' and has been prescribed by the Indian Standards Institution. It has therefore been used in the present study of the compressor performance. In this method the compressor is installed in a complete refrigerating circuit and Calorimetric measurements are made at the evaporator to determine

*Note that the definition of performance factor given in the ASHRAE guide and Data Book is the reciprocal of that given here.

the exact refrigerating effect. The refrigerating capacities of the compressor can be determined at the various operating evaporator temperatures. As far as the present author is aware, no practical performance curves for compressors are available in literature.

Following the run round the cycle method the compressor should be put in a refrigeration circuit and the results exhibited as a graph showing the variation of capacities at different evaporator temperatures.

1.2 INDIAN STANDARD

According to the Indian Standards Institution the testing of refrigerant compressor should be done in two ways independent of each other. The first is known as the Principal Test, and the second the Confirmatory Test. It prescribes three different types of principal tests and five methods of confirmatory tests.

1.2.1 Method A : Secondary Fluid Calorimeter In Suction Line : Fig 1.1(a)

In this method the low pressure vapour in the calorimeter (evaporator) is indirectly heated by vapour of a secondary fluid. This secondary fluid is in turn heated by an electrical heating device such that the heat input can be measured. The calorimeter (evaporator) is a thermally insulated vessel, the outlet of which is connected to the compressor suction. The temperatures and pressures at different points are observed.

In this method the calorimeter is complicated in construction. Further, as the vapour of the secondary fluid obtained would heat the cooling coils, accuracy of data is rather hard to obtain, due to insulation problem.

1.2.2 Method B : Dry System Refrigerant Calorimeter In Suction Line : Fig.- 1.1 (b)

In this method too, as in Method A, the liquid refrigerant in the cooling coil is indirectly heated by a second fluid which enters the calorimeter (evaporator) at a higher temperature and leaves at a lower temperature. By knowing the mass flow rate and specific heat of the secondary fluid, the heat exchanged can be calculated. The calorimeter (evaporator) is a thermally insulated vessel. The pressures and temperatures at different points in the circuit are measured.

The calorimeter in this method too, is a complicated in construction. Besides, it requires a constant temperature device and a flow control device. The heating capacity of which should be sufficient to provide the require. energy exchange in thecalorimeter. The method also depends on the accuracy with which the specific heat of the secondary fluid is known.

1.2.3 Method C : Flooded System Refrigerant Calorimeter In The Suction Line : Fig. 1.1 (c)

This method is known as Method B in the Indian Standards Institution [4].

In this method the calorimeter (evaporator) is kept flooded with liquid refrigerant upto a definite level and an electrical heating rod is kept immersed in the liquid refrigerant. Thus there is a direct heating of the liquid refrigerant. by the heater rods. The calorimeter should be thermally insulated. In this method the calorimeter is very simple in fabrication. Besides, the refrigerating capacity can be measured with greater accuracy as compared with the previous two methods, since all of the electrical input goes into the heating of the refrigerant. Hence this method has been selected for the present work.

The evaporator is utilized as a calorimeter and the energy needed to vapourize the refrigerant is determined by using heater rods, the electrical input to which may be regulated. The flow of the refrigerant is controlled by a hand regulator or constant pressure valve located close to the calorimeter.

1.2.3.1 Calibration of The Calorimeter

The calorimeter is calibrated for the heat-leakage factor by filling it with liquid refrigerant to the normal operating level and heating it so as to raise the temperature approximately 14 °C higher than the ambient. The electrical energy input is maintained constant to within 1%. If the hourly reading of the liquid refrigerant temperature does not vary by 0.6 °C, it may be assumed that the thermal equilibrium is reached. The heat leakage factor F_1, is given by the following equation :

$$F_1 = \phi_i (t_r - t_a)$$

where

ϕ_i = Heat equivalent of the energy input per minutes

t_a = Ambient temperature

t_r = Liquid refrigerant temperature.

The ISI requires that the following additional items of information should be recorded during the test.

a) The pressure of the refrigerant vapour at the evaporator outlet;

b) Pressure of the refrigerant liquid entering the expansion valves;

c) Ambient temperature at the calorimeter and

d) Electrical input to the calorimeter.

1.2.3.2 Determination of Refrigeration Capacity

Refering to p - h diagram **Fig.** 1.1 (d), the mass flow rate of the refrigerant, as determined by the test is given by the following formula :

$$\dot{m}_f = \left[\phi_h + F_1 (t_a - t_r) \right] / (h_{g2} - h_{f2})$$

where

$\dot{m}_f$ = Mass flow rate of refrigerant (kg/min)

ϕ_h = Rate of energy input to the calorimeter (kcal/min.)

F_1 = Heat leakage factor (kcal/min. °C)

t_a = Ambient temperature (°C)

t_r = refrigerant temperature (°C)

h_{g2} = Enthalpy of the saturated vapour

h_{f2} = Enthalpy of liquid refrigerant before throttling.

The refrigerating capacity, as determined by the test is given by the following formula :

$$RF = \dot{m}_f \left(h_{g1} - h_{f1} \right)$$

where

RF = Refrigerating Capacity (kcal/min.)

h_{g1} = Enthalpy of the refrigerant vapour coming out of the evaporator (kcal/kg.)

h_{f1} = Enthalpy of the liquid refrigerant at the compressor discharge condition : (kcal/kg).

1.2.4 CONFIRMATORY TEST

The confirmatory test gives the mass flow rate of the refrigerant independent of the principal methods.

1.2.4.1 Method 1 : Refrigerant Vapour Flow Meter

In this method a calibrated orifice meter is used in the compressor discharge line or in the suction line to measure the mass flow rate. This has to be calibrated against the vapour of known density. The fabrication and calibration of the orifice meter is difficult.

1.2.4.2 Method 2 : Refrigerant Liquid Quantity and Flow Meter

In this method a calibrated liquid flow meter is used.

1.2.4.3 Method 3 : Water Cooled Condenser Method

In this method an insulated water-cooled condenser is used. By measuring the temperatures of the refrigerant vapour coming in and the temperature of the liquid going out and also the inlet and outlet temperature of the cooling water, the mass flow rate of the refrigerant vapour may be calculated.

1.2.4.4 Method 4 : Refrigerant Vapour Cooling Method

In this method the total mass flow rate is determined by condensing a portion of the vapour circulated at high pressure, and measuring its quantity and then re-evaporating in a gas-cooler where the remainder of the vapour is cooled after passing through control expansion valve.

1.2.4.5 Method 5 : Refrigerant Liquid Quantity Meter

In this method too well insulated long hollow pipes each approximately 1.25 m long fitted with gauge glasses covering nearly the entire length are used, as shown in FIG 1.2. This is placed after the condenser. The liquid from the condenser comes to a header from which two lines are connected one to each metering vessel through a shut-off valve. Similarly the outlets of both the metering vessels combine at a header after passing through a valve. The fluid then finds its passage to expansion valve.

The apparatus is simple in fabrication and gives reasonably accurate results. Hence this method has been selected as a confirmatory test for the present work.

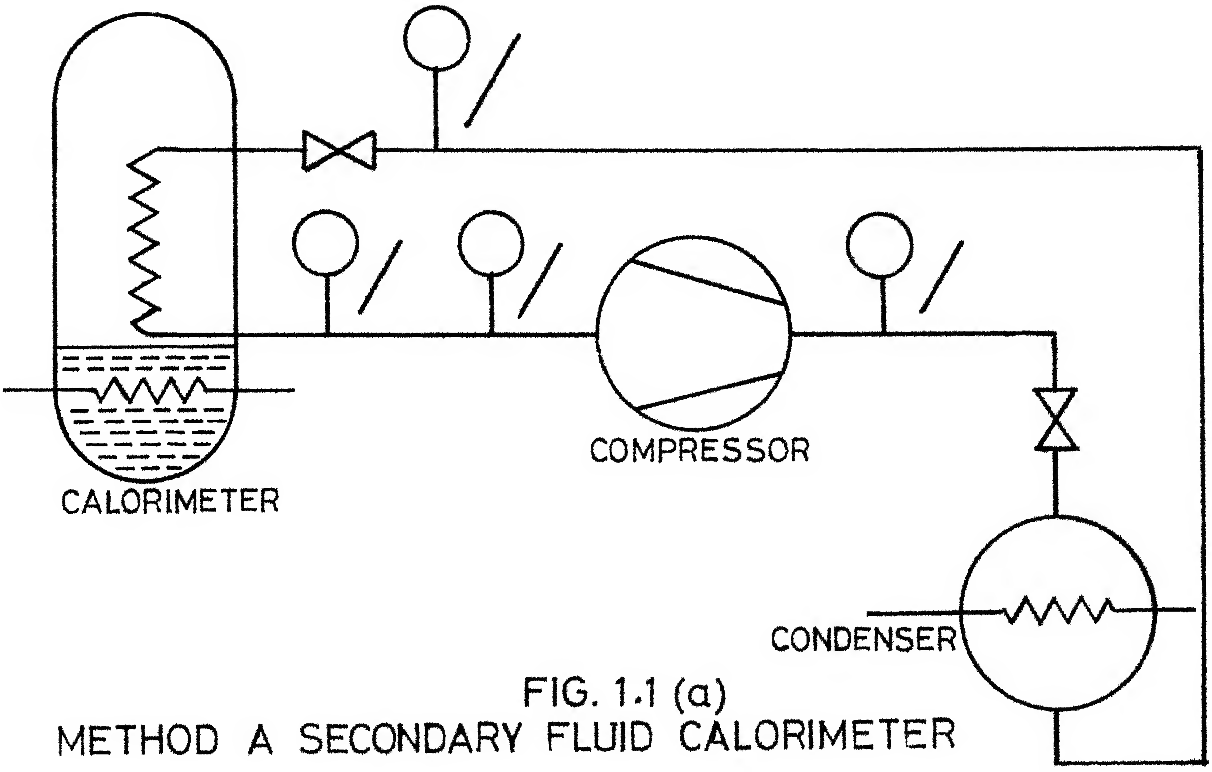


FIG. 1.1 (a)
METHOD A SECONDARY FLUID CALORIMETER

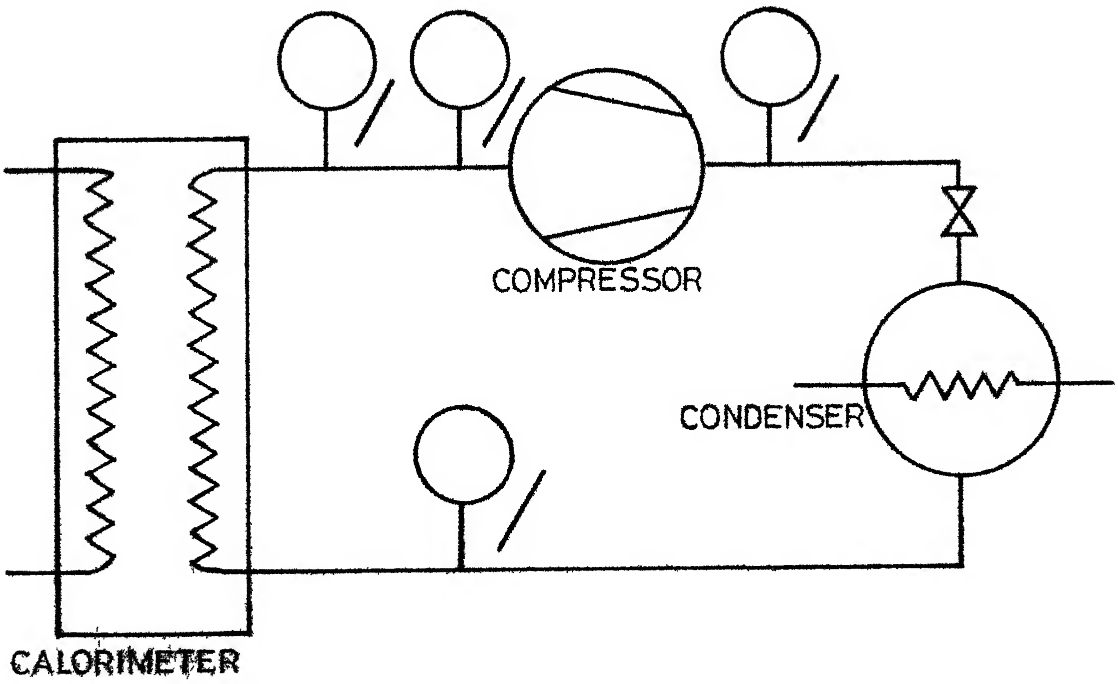


FIG. 1.1 (b)
METHOD B DRY SYSTEM REFRIGERANT CALORIMETER

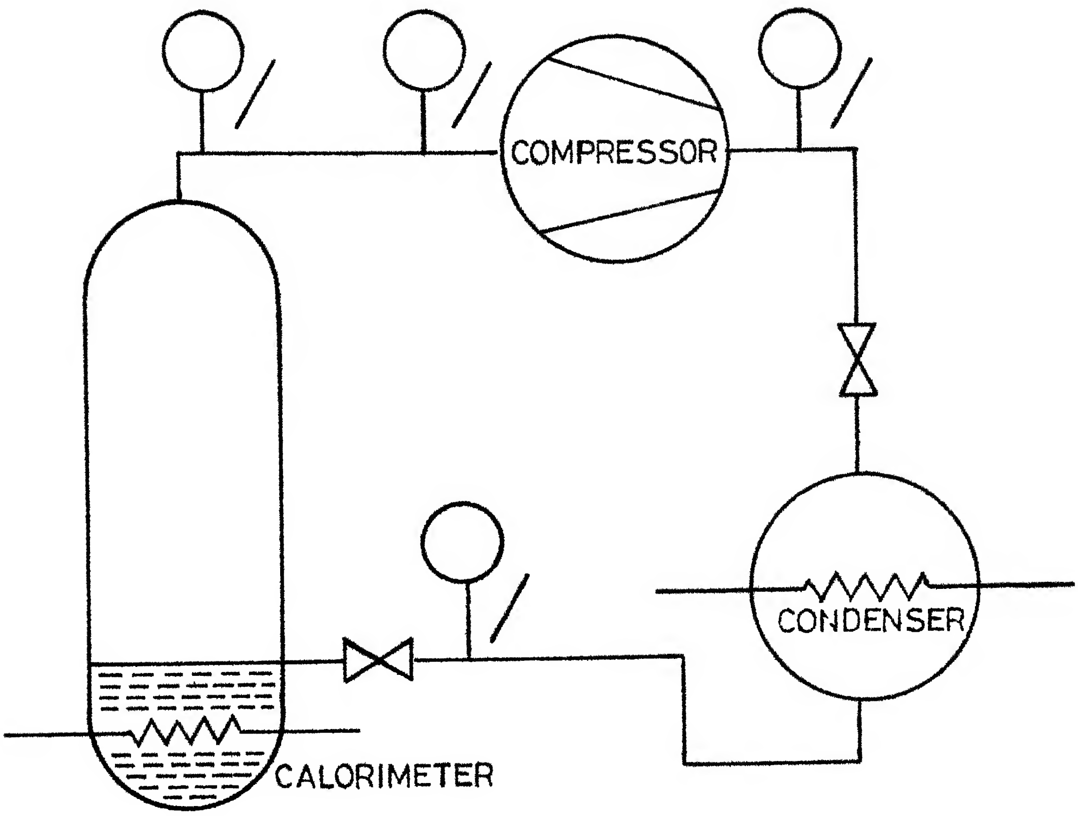


FIG. 1.1 (c)
METHOD C FLOODED SYSTEM REFRIGERANT

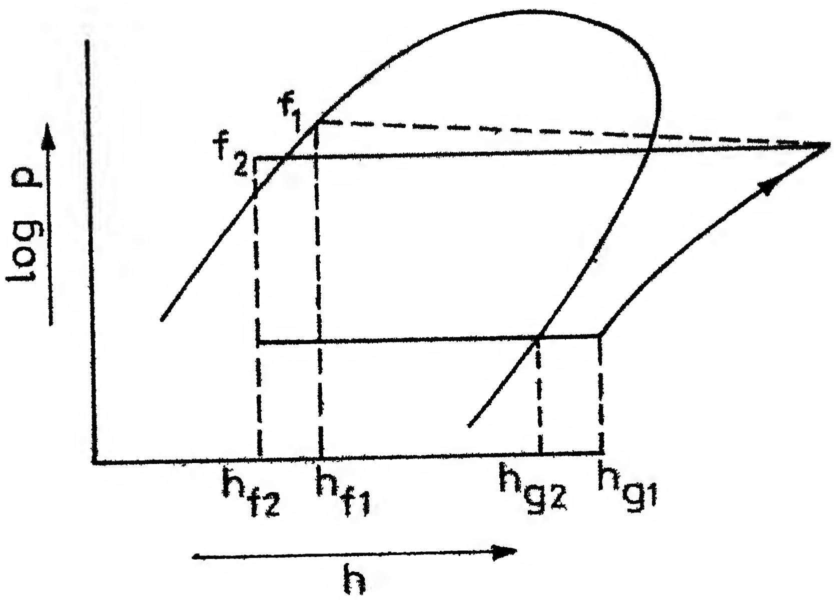


FIG. 1.1 (d)
P-h DIAGRAM

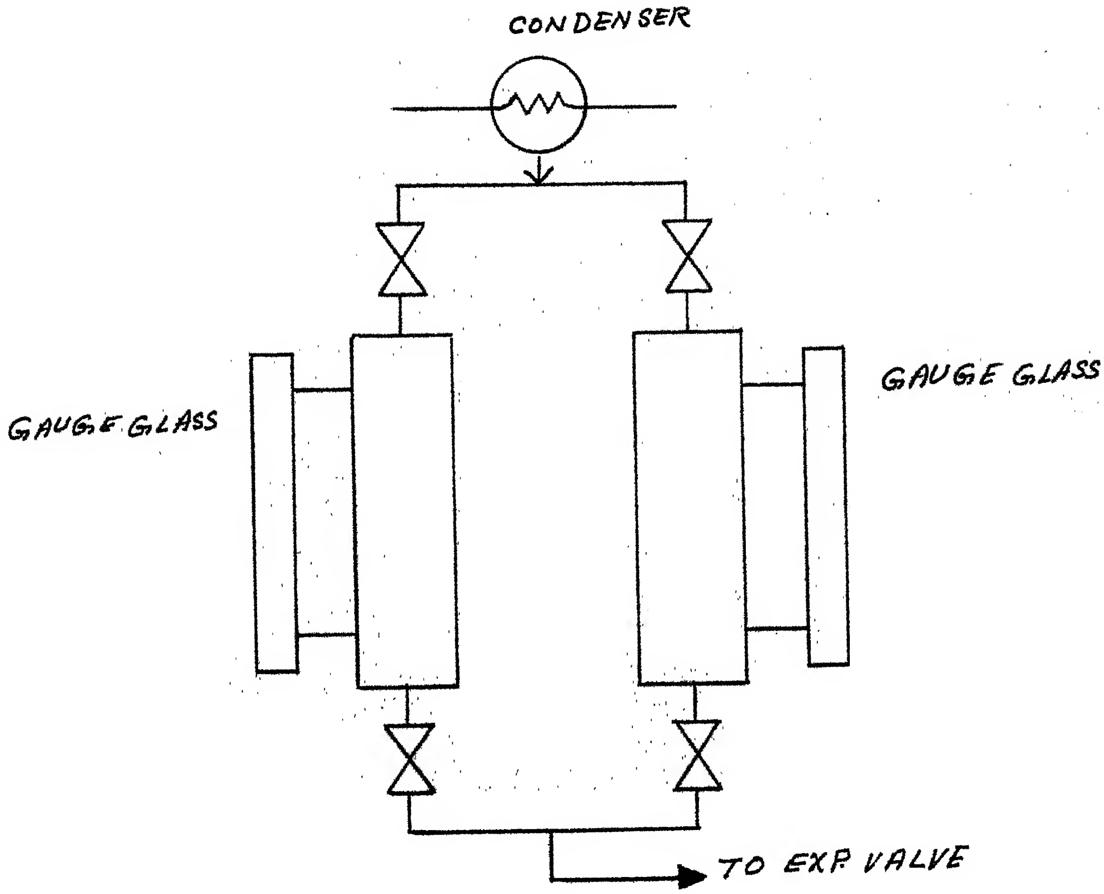


FIG.1.2 REFRIGERANT FLOW METER

CHAPTER 2

2.1 APPARATUS

A schematic diagram of the plant set-up for the present work is shown in Figs. 2.1 and 2.2 and photographs are shown in Figs. 2.3 and 2.4. As the compressor is tested during the 'run round the cycle test' the entire plant has been set-up using all the accessories of a simple vapour compression refrigeration system. The compressor is tested for its refrigeration capacities at various evaporator temperatures. In addition a flow-meter has been used in the system in accordance with ISI specifications to confirm the mass flow rate of the liquid refrigerant in the system.

The following are the equipment used in the plant set-up :

2.1.1 Compressor

A positive displacement compressor A, manufactured by Freezeking of India Ltd., is tested in the present work and has been used to compress the low pressure refrigerant vapour to condenser pressure. At both the suction and discharge ends of the compressor; needle valves have been included to regulate the flow through the system. Pre-calibrated copper constantum thermocouples of 24 BSW gauge have been inserted at both ends of the compressor (in the suction and discharge lines), to measure the refrigerant suction and discharge temperatures.

2.1.2 Oil Separator

In the Compressor the refrigerant comes in contact with lubricating oil in the crank-case. So the refrigerant flows out mixed with the oil after compression. Refrigerant 12, used in the present work is completely miscible in oil so that no oil seperator is usually used in R-12 systems. However, in the present work as a precautionary measure, an oil seperator, B, is included in the circuit to ensure that even tiny oil droplets are not carried by the refrigerant into the condenser. (If the oil is not seperated, it may collect in the condenser or evaporator tubes and adversely affect the heat transfer characteristics).

2.1.3 Condenser

A two pass water-cooled condenser D of nominal capacity 3 tons, manufactured by American Refrigeration Company India, is used to condense the compressed refrigerant vapour into a liquid. The condenser is over-sized so that the liquid refrigerant at the condenser exit may be slightly sub-cooled.

Cooling water is passed to the condenser through a rotameter to measure its flow rate. Pressure gauges and thermocouples are fitted in the refrigerant line at the condenser inlet and outlet respectively.

2.1.4 Drier and Filter

A drier filter combination, B, consisting of a silica-gel and calcium chloride, has been used in the circuit before the refrigerant enters the expansion valve or needle valve for throttling. Any suspended impurities will be arrested in the silica-gel of the filter and the moisture, if any, will be removed in the drier. Both the suspended impurities and the moisture are detrimental to the refrigeration system and specially for the refrigeration system which has to go to 0 °C or lower, since the small opening in the expansion valve may be choked and the flow stopped. The plant may have to stop due to suction pressure becoming too low.

2.1.5 Expansion Valve and Needle Valve

A two-ton thermometric expansion valve has been fitted into the circuit as shown. In addition a needle type of valve has been fitted in series with thermostatic valve to obtain the desired variation in pressure in the evaporator.

2.1.6 Calorimeter (Evaporator)

The purpose of the calorimeter is to find the refrigerating capacity of the compressor at the different test conditions.

The calorimeter, **Fig.** 2.6, is made in the form of cylindrical shell 22.9 cm. diameter cut of brass sheet 6 mm thick. The calorimeter is large enough to accumulate sufficient

refrigerant liquid to act as a liquid receiver as well since no liquid receiver has been provided anywhere else in the system.

The liquid refrigerant collected inside the calorimeter is heated by using five electric heater rods, each 100 cm long and 1 kw heating capacity, inserted into mild steel, 1.25 cm nominal internal diameter tubes, brazed to both the end plates of the calorimeter. The liquid refrigerant level is so maintained that pipes carrying the heater rods are always immersed in it.

The calorimeter is insulated with 7.5 cm thick fibre-glass insulation all round and also at the end plates. The pipes connecting the expansion valve to to the calorimeter are also similarly insulated.

Before being fitted in the plant, the calorimeter is tested with air at a pressure of 140 PSIG (App. 9.5 atm). Leakages are detected by using concentrated soap solutions. The leaking joints are all brazed and tested again. This process is continued untill no leakage is seen.

The method of calibrating the calorimeter is discussed in Appendix A-2.

2.2 INSTRUMENTATION

2.2.1 Flow Meter (Condenser Cooling Water)

For measuring the water flow rate in the condenser a rotameter of the capacity 0-9 gpm (with least count 0.2 gpm) is used. The calibration process for the flow meter is discussed in Appendix A-1.1.

2.2.2 Flow Meter (Liquid Refrigerant)

For liquid refrigerant flow measurement a flow meter has been fabricated according to ISI specifications. It consists of two 7.5 cm nominal internal diameter mild steel pipes approximately 1.25 m long. (This dia and length has been chosen for the flow meter such that liquid refrigerant may collect and rise in level of minimum 150 mm in 2 minutes). On each pipe, two gauge glasses are fitted as shown in Fig 2.5 extending over the entire length of the meter. The ends of the pipe are flanged, to provide connection to the condenser and to the expansion valve. The whole lengths of the pipes is insulated by 5 cm thick fibre-glass insulation. At the outlet header mentioned earlier, a thermocouple and a pressure gauge have been inserted to make appropriate measurements.

This meter will measure the volume flow rate and the readings will be multiplied by density to give the mass flow rate. The details of construction are shown in the figure. One pressure equalizer is fitted to maintain both

the pipes of the flow meter in communication with each other. Two shut off valves, one each at the inlet and outlet of each pipe are used. The reading of the flow meter is taken in the pipe in which the level is rising and not in the pipe in which the level is falling.

The details of calibration of the flow meter are given in Appendix A-1.2.

2.3 TEMPERATURE MEASUREMENT

The temperature of the refrigerant are measured at different points by means of a 24 - BSW gauge copper constantum thermocouple. These are inserted into wells made by brazing nearly 5 cm long 6mm dia copper tube at the point where the temperatures are to be measured (Fig.2.7) one swage lock tee (imported) having three nuts is fitted to this piece of 6 mm dia copper tube. At this point the thermocouple bead is inserted. The lead wires are taken to the selector switch and through the selector switch the connection goes to the potentiometer (Least count .005 mv).

The details of calibration are given in Appendix A-1.3.

2.4 PRESSURE MEASUREMENTS

All pressures are measured using Bourdon-tube pressure gauges. The pressure gauge used at the evaporator inlet and outlet is of 0-100 PSIG range with a least count of 0.05 Psi. A pressure gauge 0 - 400 PSIG range (least count 2.5 Psi) is

used to measure the pressure at this compressor discharge and condenser inlet. Another gauge, 0 - 300 PSIG range (least count 1.25 Psi) is used to measure the pressure at the condenser outlet and the flow meter outlet.

All these pressure gauges are calibrated against a dead-weight pressure gauge. The calibration procedure is discussed in Appendix A-1.4.

2.5 ENERGY METER

(i) A three phase energy meter is used to measure the electrical energy input to the motor driving the compressor. The procedure used in calibrating it is discussed in Appendix A-1.5.

(ii) A single phase energy meter is used to measure the energy input to the calorimeter. The calibration procedure is discussed in Appendix A-1.6.

2.6 EXPERIMENTAL PROCEDURE

The compressor is started and the expansion valve or the needle valve is adjusted to give a definite flow rate. Then the heating capacity of the calorimeter is adjusted with a variac. When it is seen that the pressure readings have remained constant for about 15 minutes, the following observations are made :

(i) The pressure gauge readings at the various indicated points.

(ii) The length of the time needed for 10 revolutions of the three-phase energy meter (motor input) and also for 10 revolutions of the single-phase energy meter (evaporator input) are determined by using a hand held stopwatch.

(iii) The inlet valve of the pipe No. 1 (FL1) of the flow meter is closed, the outlet valve opened while the inlet valve of pipe No. 2 (FL2) of the flow meter is opened and the outlet closed. The rise in level in the gauge glass of pipe No. 2 is observed over a period of 2 minutes (approximately).

(iv) The thermocouple readings at various points are taken, using a potentiometer.

(v) The water flow rate to the condenser is noted using the rotameter.

This completes one set of readings. The expansion valve is now adjusted either to increase or decrease the flow rate and consequently a different pressure and temperature are produced in the calorimeter (evaporator). Then the readings are taken for the second set in the same way as described above and the process repeated untill several sets of readings are obtained.

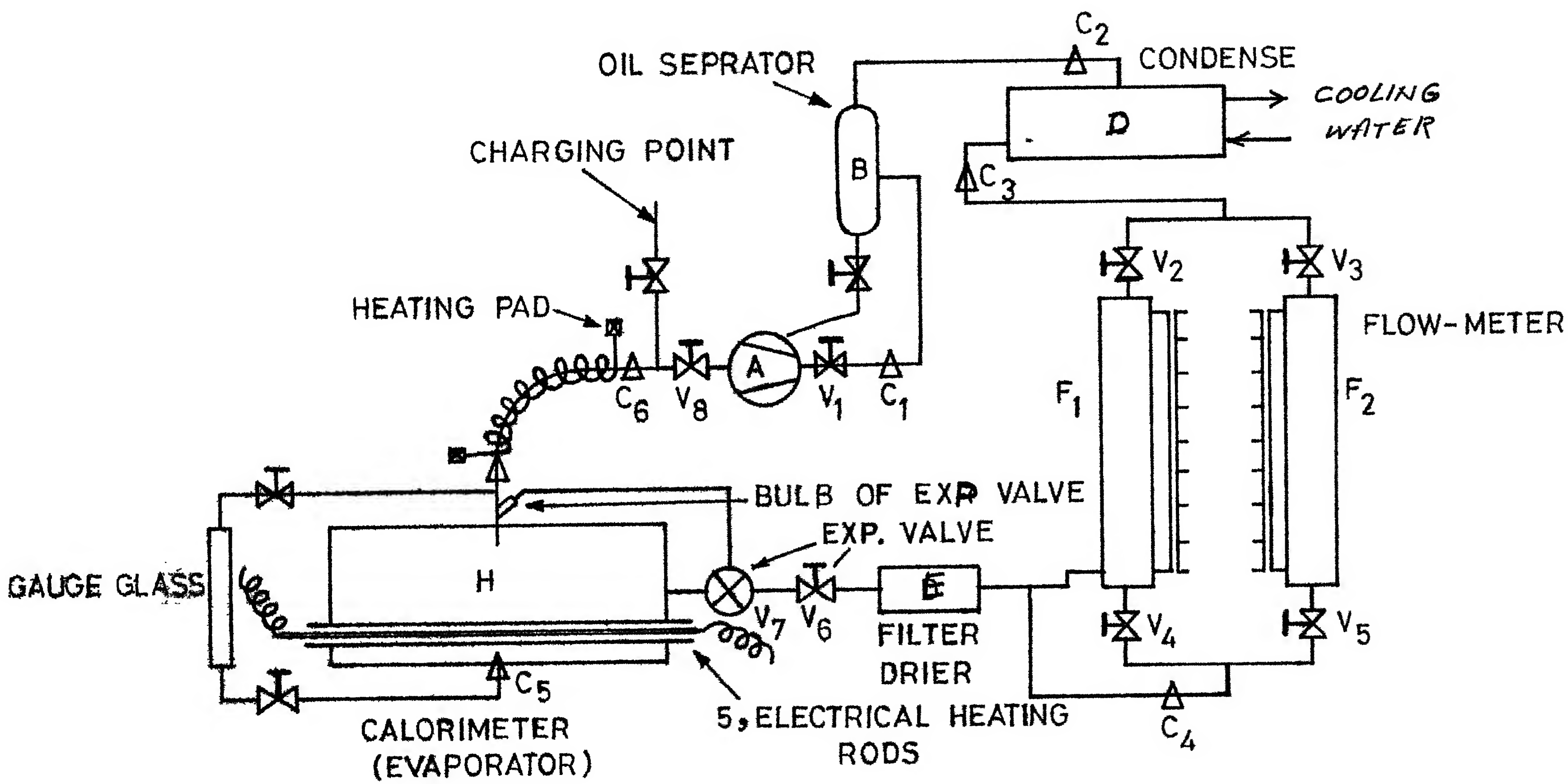


FIG. 2.1 SCHEMATIC DIAGRAM OF PLANT LAY - OUT

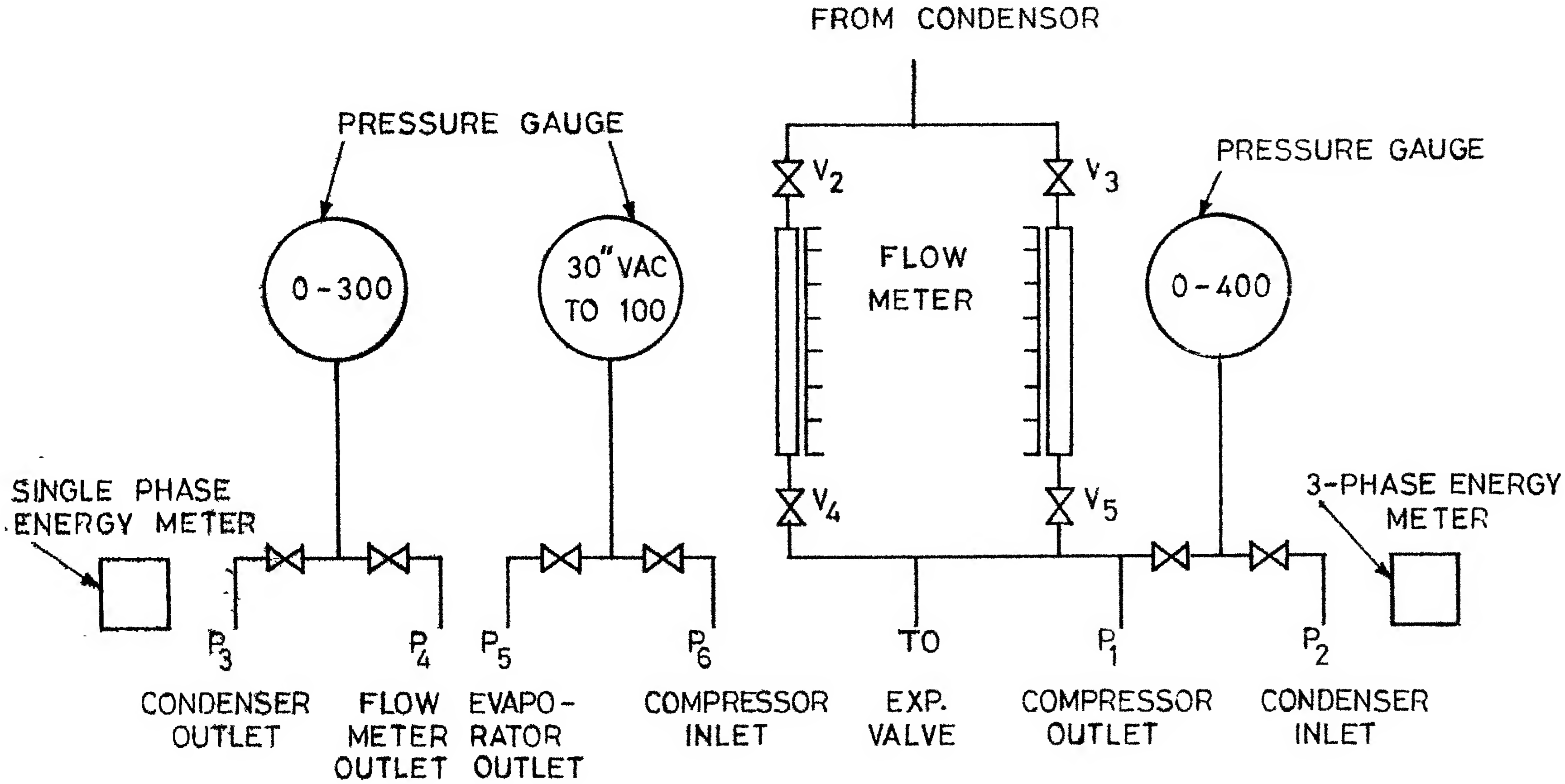
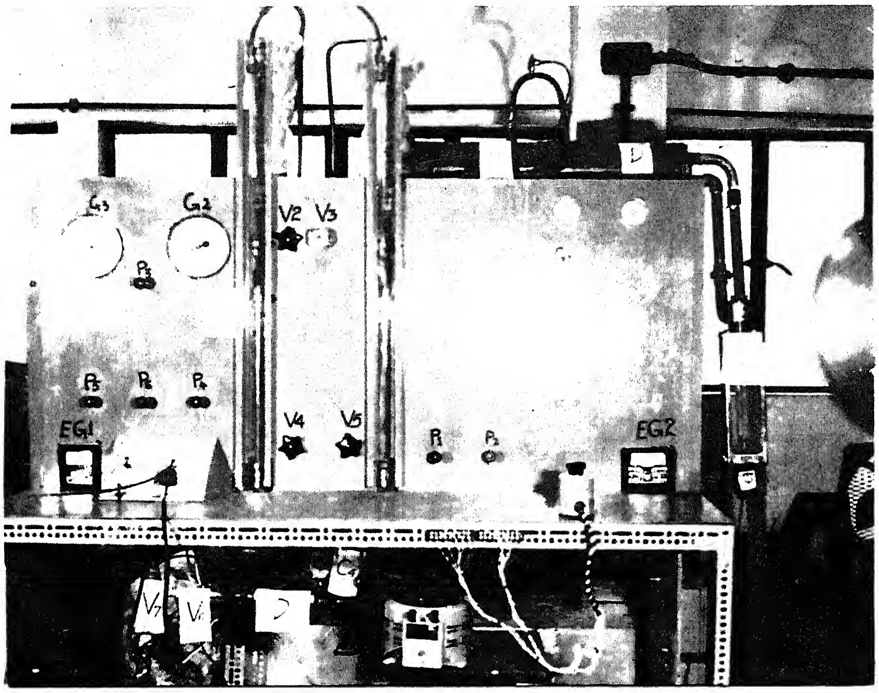
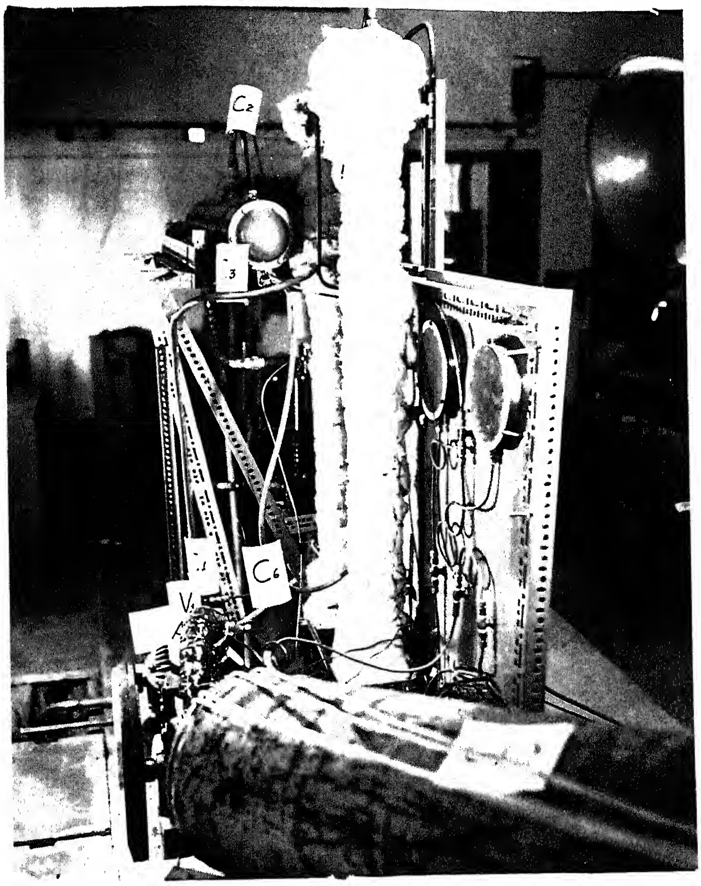


FIG. 2.2: SCHEMATIC DIAGRAM OF CONTROL-PANEL


G2
V2
V3
V4
V5
EG1
EG2


C2
C6

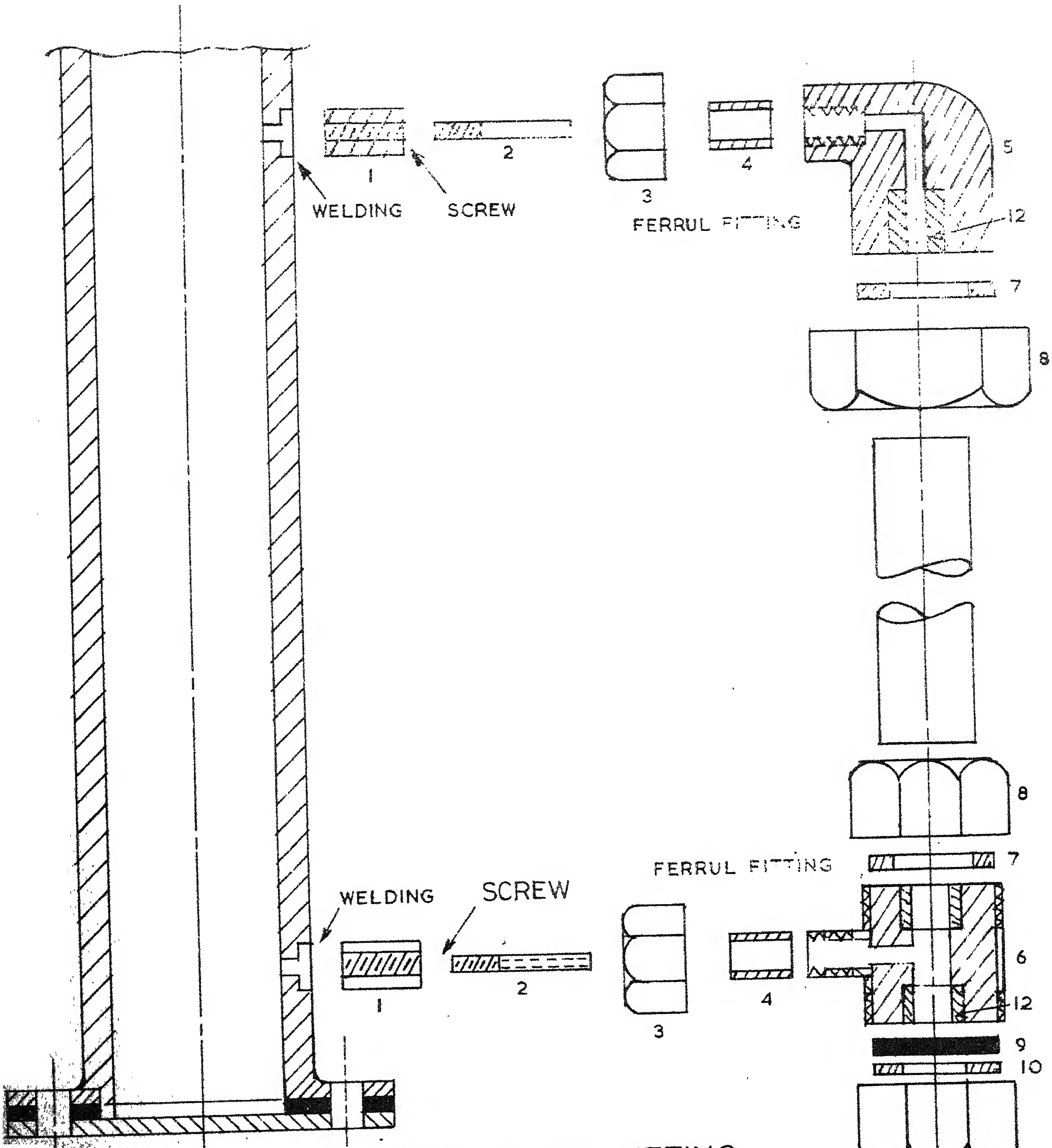


FIG.2.5 DETAILS OF GAUGE GLASS FITTING

1 1/4" M S SOCKET
2 1/2" COPPER TUBE (THICK WALL) NEEPLE
3 1/2" HEXAGONAL NUT (BRASS)
4 FERRUL
5 1/2" ELBOW (BRASS)
7 SYNTHETIC RUBBER O RING
10 M . S . RING
8 & 9 BRASS 1/2" HEXAGONAL NUT
12 TAFLEN RING

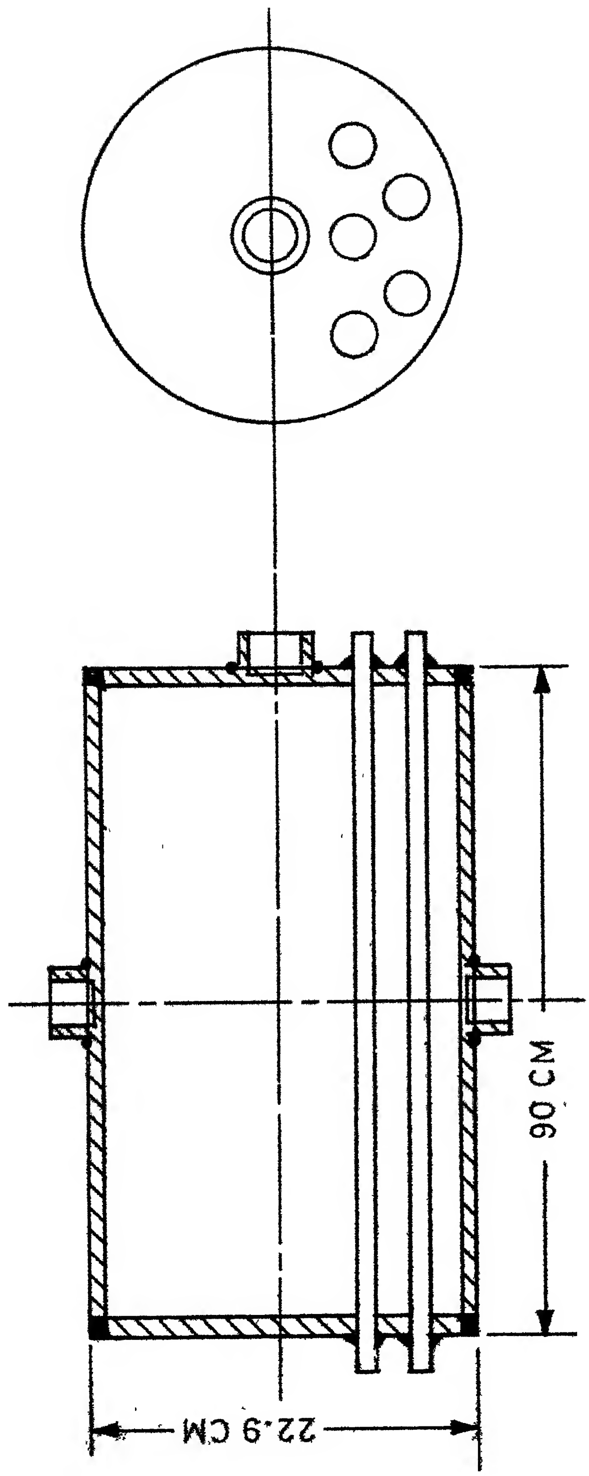


FIG. 2.6 DETAILS OF CALORIMETER CONSTRUCTION

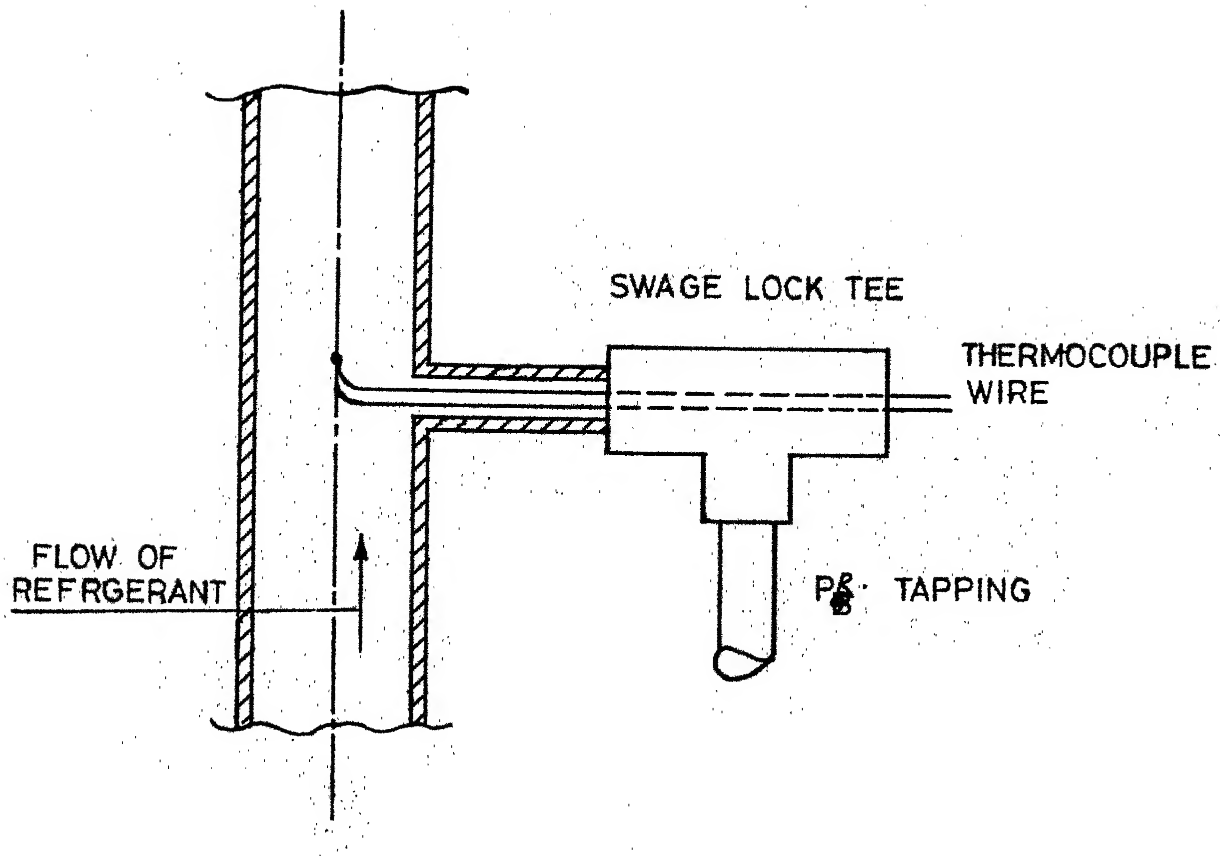


FIG. 2.7 DETAILS OF THERMOCOUPLE AND PRESSURE TAPPINGS.

CHAPTER 3

3.1 DATA REDUCTION

The needle valve or the expansion valve is set at a particular opening so as to give a definite flow rate which will result in a definite suction and discharge pressure. The plant is run for some time to determine whether the pressure readings are constant.

Then the following readings are noted and the data reduced as follows :

The Energy Meter (EG 1) is read with a stop watch and time for 10 revolutions of the disc noted. Then from the calibration chart, the wattage kW is read against this RPM of the disc. Let this wattage be kW_1. Then $kW_1 * \frac{860}{60}$ or $kW_1 * 14.3$ kcal/minute is the energy put to the calorimeter.

By opening the respective valves, the following pressures are read. The correct values of the pressure is read using the calibration curves, and are also listed below :

		Reading of the pressure gage	Correct pressure
(i)	Compressor discharge pressure	P_1	PC_1
(ii)	Condenser Inlet Pressure	P_2	PC_2
(iii)	Condenser outlet Pressure	P_3	PC_3
(iv)	Pressure of the liquid refrigerant before it enters the expansion valve.	P_4	PC_4
(v)	Evaporator outlet pressure	P_5	PC_5

The potentiometer readings are noted, mV, with reference junction in atmosphere.

		Millivolt	°C
(i)	Compressor discharge temperature	TC_1	TCC_1
(ii)	Condenser Inlet temperature	TC_2	TCC_2
(iii)	Condenser outlet temperature	TC_3	TCC_3
(iv)	Temperature before expansion valve	TC_4	TCC_4
(v)	Temperature just after the expansion valve	TC_5	TCC_5
(vi)	Temperature at the suction of the compressor	TC_6	TCC_6

The water flow rate of the condenser is noted from the rotameter say, RM, gpm. From the calibration chart, the rate of flow is kg/min., RMC, is determined.

F_1 is the heat leakage factor of the calorimeter in kcal/min. °C obtained from its calibration given in Appendix A-2.

3.2 DETERMINATION OF REFRIGERATION CAPACITY

The mass flow rate of the refrigerant is given by the equation

$$\dot{m}_f = \left[\phi_h + F_1 (t_a - t_r)\right] / (h_{g1} - h_{f2})$$

where

$\dot{m}_f$ = mass flow rate of refrigerant (kg/min)

ϕ_h = energy input to the calorimeter (kW_1 * 14.3 kcal/min.)

F_1 = Heat leakage factor of the calorimeter (kcal/min)

t_a = Ambient temperature (°C)

t_r = Refrigerant temperature in the evaporator (which is TCC_5).

h_{g1} = Enthalpy of the refrigerant leaving the calorimeter (kcal/kg)

h_{f2} = Enthalpy of refrigerant entering the calorimeter (kcal/kg)

and the refrigeration capacity RF by the equation :

RF = Refrigeration capacity (kcal/min) $= \dot{m}_{r} \{ h_{g1} - h_{f1} \}$

3.3 DETERMINATION ENERGY INPUT TO MOTOR

One energy meter (EG2) is fitted across the motor driving the compressor. Knowing the speed of rotation of the disc in RPM, the power input to the motor, kW_2 is obtained.

Motor losses determination is given in the Appendix A-4.

The motor losses calculated on the basis of kW_2 is $\left[(kW_2/3.5)^2 (600) + 300 \right] /1000$. Let it be L_2 (kW).

Net power output of this motor = $(kW_2 - L_2) = KWC_2$ (kW)

Assuming a 95% transmission efficiency (double Vee groove belt), the power input to the compressor is KWC_2 *.95. Let it be equal to $SKWC_2$.

3.4 DETERMINATION OF CO-EFFICIENT OF PERFORMANCE

The co-efficient of performance, COP, is given as follows :

COP = Refrigeration effect/Power Input to the shaft

The heat equivalent to $SKWC_2$ is equal to $(SKWC_2)$ (14.3) k cal per minute.

$$COP = (RF)/(SKWC_2)\ (14.3)$$

3.5 DETERMINATION OF RELATIVE COP

The isentropic work done by the compressor in compressing the saturated vapour at the outlet of the evaporator to the condenser pressure, is seen from the p-h chart of R-12. Let it be ϕ_s.

$$(COP)_S = \phi_s / (h_{g2} - h_{f1})$$

The relative co-efficient of performance $(COP)_R$, is the ratio of (COP) to $(COP)_S$.

3.6 DETERMINATION OF SPECIFIC CAPACITY

The specific capacity, $(RF)_s$, is the refrigeration effect per kW of energy input to the shaft of the compressor per hour.

$$(RF)_s = (RF)\ (60) / (SKWC_2) = (COP)\ (860.0)$$

CHAPTER 4

4.1 RESULTS AND DISCUSSIONS

The performance curve of the reciprocating compressor (Model FK - 450, manufactured by Freeze King India Ltd.) has been shown in Fig. 4.1. The refrigeration capacity per hour is calculated at the various evaporator temperatures. While calculating the refrigeration capacity, the increase of capacity due to sub-cooling is not accounted for because as per ISI specifications, this increase in capacity should not be credited to the compressor. The computations provide the following parameters, the details of which are given in the Chapter 3 and a sample calculation in Appendix A-3.

1. The refrigeration capacity, RF (kcal/min.)
2. The energy input to the motor driving the compressor, kW_2,
3. The co-efficient of performance and,
4. The mass flow rate of the refrigerant.

The plot of refrigeration capacity vs the evaporator temperature has been shown in the Fig. 4.1. It shows that the capacity increases with increasing evaporator temperature. The curve tends to become steeper as the evaporator temperature increases. The curve shows that the rate of increase in capacity (with respect to evaporator temperature) is comparatively low at low evaporator temperatures and larger at higher temperatures.

This behaviour of the capacity can be theoretically justified as follows :

The refrigeration capacity is the product of the mass flow rate and the difference of the refrigerant enthalpies at the evaporator outlet and inlet conditions. The thermodynamic behaviour of the refrigerant, R-12, is such that the differences of enthalpies stated above tends to diminish with increase of pressure in the evaporator. Further as the evaporator temperature increases the specific volume of the refrigerant at the compressor inlet decreases. Hence, as the volume swept per hour by the compressor is fixed, the mass flow rate of the refrigerant increases with the increasing evaporator pressures. The increase in mass flow rate is much more than the decrease in the difference of enthalpies between the evaporator outlet and inlet conditions (at higher evaporator pressure). Thus the refrigeration capacity should increase with increasing evaporator pressure.

The plot of the COP with evaporator temperature is shown in Fig. 4.2. It is seen that the COP is 1.13 at $-28^{\circ}C$ and 2.76 at $0^{\circ}C$. The plot exhibits that the COP increases very smoothly between these two limits. It is evident from T-S diagram for R-12 that for a given condensor temperature, increase of evaporator pressure decreases the compressor work as well as the refrigeration effect. However the changes are such that the ratio of refrigeration effect to the

compressor work increases with increased evaporator pressures. The thermodynamics of the refrigerant dictates that the COP should increase with increasing evaporator pressure. Thus we find that the behaviour of COP curve obtained in the present investigation is well in accordance with theoretical considerations.

The energy input to the compressor has been plotted against the evaporator temperature in Fig. 4.3. The figure exhibits that the energy required to drive the compressor is relatively smaller at low evaporator temperature than at high evaporator temperature. This behaviour is also supported by the theory explained for the COP behaviour above.

BENDIXEN [2] has tested a few refrigerant compressors to show that counter flow compressors were superior to uniflow compressors. The theory behind this is that during the compression stroke liquid drops settle on the cold piston top in the uniflow compressor, and that this liquid evaporates during the suction stroke and that there is a certain uniformity of the wall temperature in the cylinders of the counter flow compressor resulting in lower wall loss. According to the author, if this theory is correct; the volumetric efficiency of the counter flow compressor should be higher than that of the uniflow compressor. Further the less the superheat is, the higher volumetric efficiency should be relatively.

Through the plot of volumetric efficiency versus suction superheat and also through the plot of volumetric efficiency versus compression ratio, the author concludes that there is a difference of only a few percent of the volumetric efficiency between these two types of compressor and down to 5 °C., there is no appreciable fall of the volumetric efficiency for the uniflow compressor than for the counter flow compressor. Hence the author has proved that it is no importance of the volumetric efficiency of a refrigerant compressor, if the suction gas is led into the cylinder at its coldest or hottest part.

CHAIKOVSKY, SHMIGLYA and SAVKOV [3] has performed experiments on compressors and have plotted curves of refrigerating capacity versus evaporator temperature. For FV-12 compressor having t_c = 35 °C and C = 1.94 %, and for FU - 8 compressors, Fig. 4.4, the results show that they have gone to a minimum evaporator temperature of - 15 °C. However in the present work minimum of - 28 °C evaporator temperature has been obtained. The figure of experimental set-up is shown in Fig. 4.8.

The method of test is not clear in the above mentioned article. These curves may be compared with the curves of the present work. It is clear that the performance of the compressors are similar to those obtained in the present work inspite of the differences in make and methods of testing employed.

The heat removed in the condenser has been plotted in Fig. 4.5 against the evaporator temperature. The curves show rising trend at higher temperatures. We have already seen that the energy input to the compressor shaft and the refrigerating capacity increase with increase of evaporator temperatures. Hence the energy withdrawn at the condenser should increase with the increase of evaporator temperature, as it is equal to the sum of the refrigerating effect and the energy input to the compressor.

The mass flow rate has been plotted against the evaporator temperature in the Fig. 4.6. This gives the idea of volumetric efficiency of the compressor. From the previous discussions, we know that the mass flow rate plays an important role in the determination of parameters like refrigerating capacity and COP.

The theoretical co-efficient of performance of a simple vapour compression refrigeration cycle between the evaporator and condenser pressure has been calculated assuming the refrigerant vapour at the outlet of the evaporator to be dry saturated and isentropic compressor work. Then the relative co-efficient of performance is plotted in the Fig. 4.7 against the evaporator temperature. The relative co-efficient of performance increases from 30% at -28 °C to ~~42%~~ 43·5% at 0 °C and the rate of increase is very smooth.

4.2 ERROR ANALYSIS

The error analysis is an important step to be carried out for an estimate of errors involved in various measured quantities in the present investigation. The accuracies of the final results depend upon the various errors encountered during the measurement of different parameters involved. The error analysis presented here gives an estimate of errors in the maximum possible range and the actual error is expected to lie with this range.

ERRORS

1. Pressure Measurements

The pressure gauges are calibrated using dead weight pressure gauge. Hence no systematic error is expected to be present. The random error present in the pressure gauge is $\pm$ 2.54% fitted for the low pressure side and $\pm$ 2.18 % fitted for the high pressure side.

2. Temperature Measurements

The thermocouples used for the measurements of temperatures at the various points in the circuit and calibrated using a platinum resistence thermometer. The error involved is negligible as the accuracy of this thermometer is $\pm$ .005 °C. The error involved in measuring the temperature with the thermocouple using the millivolt potentiometer is $\pm$ (0.24/28) * 100 = $\pm$ 0.856% in the low pressure side and $\pm$ (0.24/60) * 100 = $\pm$ 0.4% in the high pressure side.

The ambient temperature is measured by the thermometer having error in readings $\pm$ (0.1/67) * 100 = 1.49%.

3. Water Flow Rate Measurements

No systematic error is involved in the rotameter used for the measurement of water flow rate as it has been calibrated by collecting water during a known time. The random error in this case is (0.20/4) * 100 = $\pm$ 5%.

4. Measurement of Electrical Power

The energy meter was calibrated using an ammeter a voltmeter and a hand held stop watch. The calibration curve is plotted (kW vs RPM). Thus assuming no systematic error involved in the ammeter and voltmeter, the error in the measurements is zero.

5. Measurement of Refrigerant Liquid Flow Rate

The flowmeter is calibrated by collecting demineralized water in the flowmeter. The error involved in this case is random, and less than (0.1 x 100)/20 = $\pm$ 0.5%.

6. Errors in The Estimation of h_{g1}, h_{f1}, h_{f2}

For $\pm$ 2.54% error in the measurement, the error involved in the measurement of the h_{g1} is $\pm$ (0.05/135.7) = $\pm$ 0.037.

For $\pm$ 2.18% error in the measure, the error involved in the measurement of h_{f1} and h_{f2} is (0.01/105) = $\pm$ 0.00955.

7. The Error in the Calculation of F_1

$$F_1 = \phi_i / (t_r - t_a)$$

$$(\Delta F_1/F_1) = \pm (\Delta \phi_i / \phi_i) \pm \Delta (t_r - t_a) / (t_r - t_a)$$

$$= \pm (0.24/14) * 100 = \pm 1.7\%$$

8. The error in calculation of $\dot{m}_f$

$$\dot{m}_f = \phi_h + F_1 (t_a - t_r) / (h_{g1} - h_{f2})$$

$$\Delta \dot{m}_f / \dot{m}_f = \pm \Delta F_1 / F_1 \pm \Delta (t_a - t_r) / (t_a - t_r)$$

$$\pm \Delta(h_{g1} - h_{f2}) / (h_{g1} - h_{f2})$$

$$= \pm 0.17 \pm 0.01712 \pm 0.0465 = \pm 0.08062$$

9. Error estimate in the Calculation of RF (Refrigeration Capacity).

$$RF = \dot{m}_f (h_{g1} - h_{f1})$$

$$\Delta RF/RF = \pm \Delta \dot{m}_f / \dot{m}_f \pm \Delta(h_{g1} - h_{f1}) / (h_{g1} - h_{f1})$$

$$= \pm 0.08063 + 0.0465$$

$$= \pm 0.1271$$

10. Error in the Calculation of COP

$$COP = RF/kW$$

$$\Delta COP/COP = \pm \Delta RF/RF \pm \Delta kW/kW$$

$$= \pm 0.1271$$

4.3 CONCLUSIONS

A refrigerant R-12 compressor test rig has been fabricated in accordance with ISI specifications to determine compressor performance. A test rig of this type is quite general in comparison with the other methods cited in literature and unique in India to the best of author's knowledge. A compressor, open type, manufactured by Freeze King India Ltd. has been used to determine the performance characteristics which have been exhibited in the form of curves such as refrigeration capacity versus evaporator temperature, co-efficient of performance versus evaporator temperature and kW (power input) to evaporator temperatures etc. The results obtained in the present investigation are quite general for practical applications. However a gross estimate of the error involved indicates that the results may be off by nearly $\pm 13\%$. A better design of the evaporator, in respect of uniform distribution of the heater rods and some more insulation, along with better instrumentation would definitely bring down the error to a close estimate.

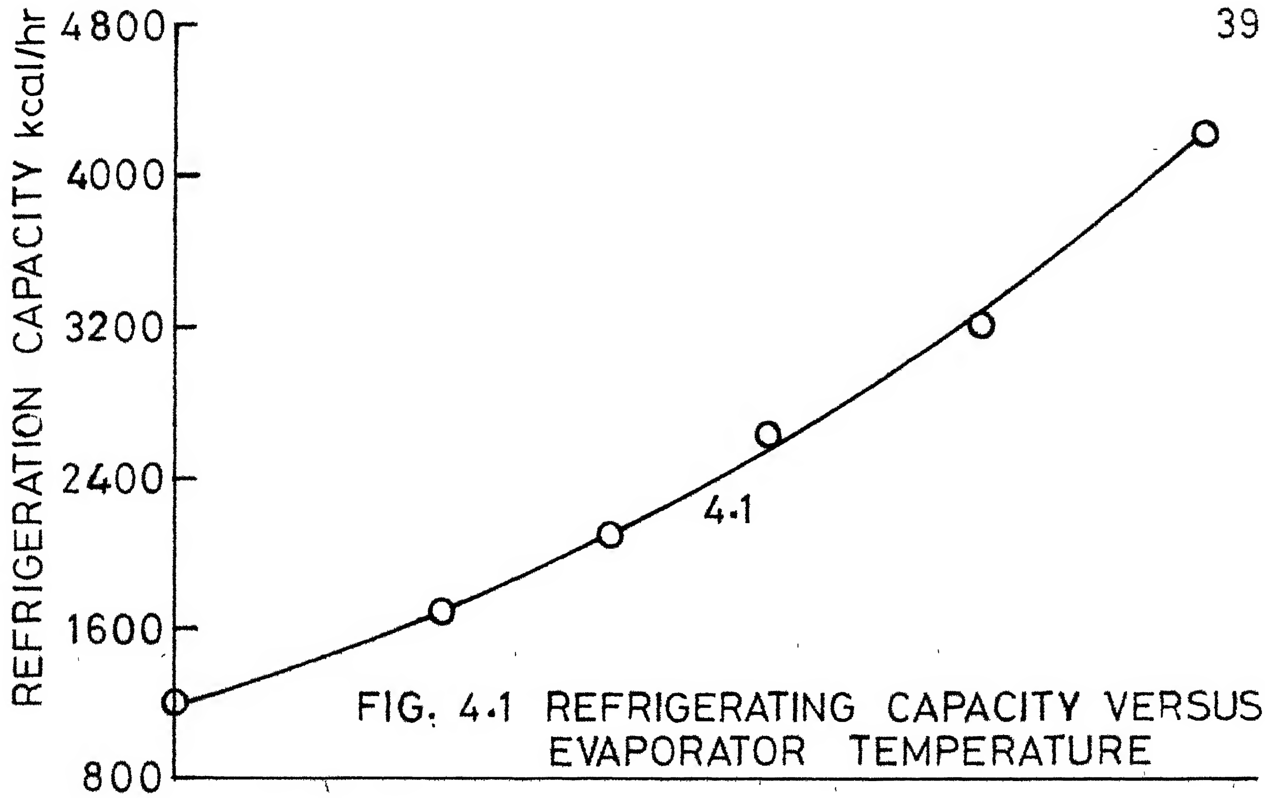


FIG. 4.1 REFRIGERATING CAPACITY VERSUS EVAPORATOR TEMPERATURE

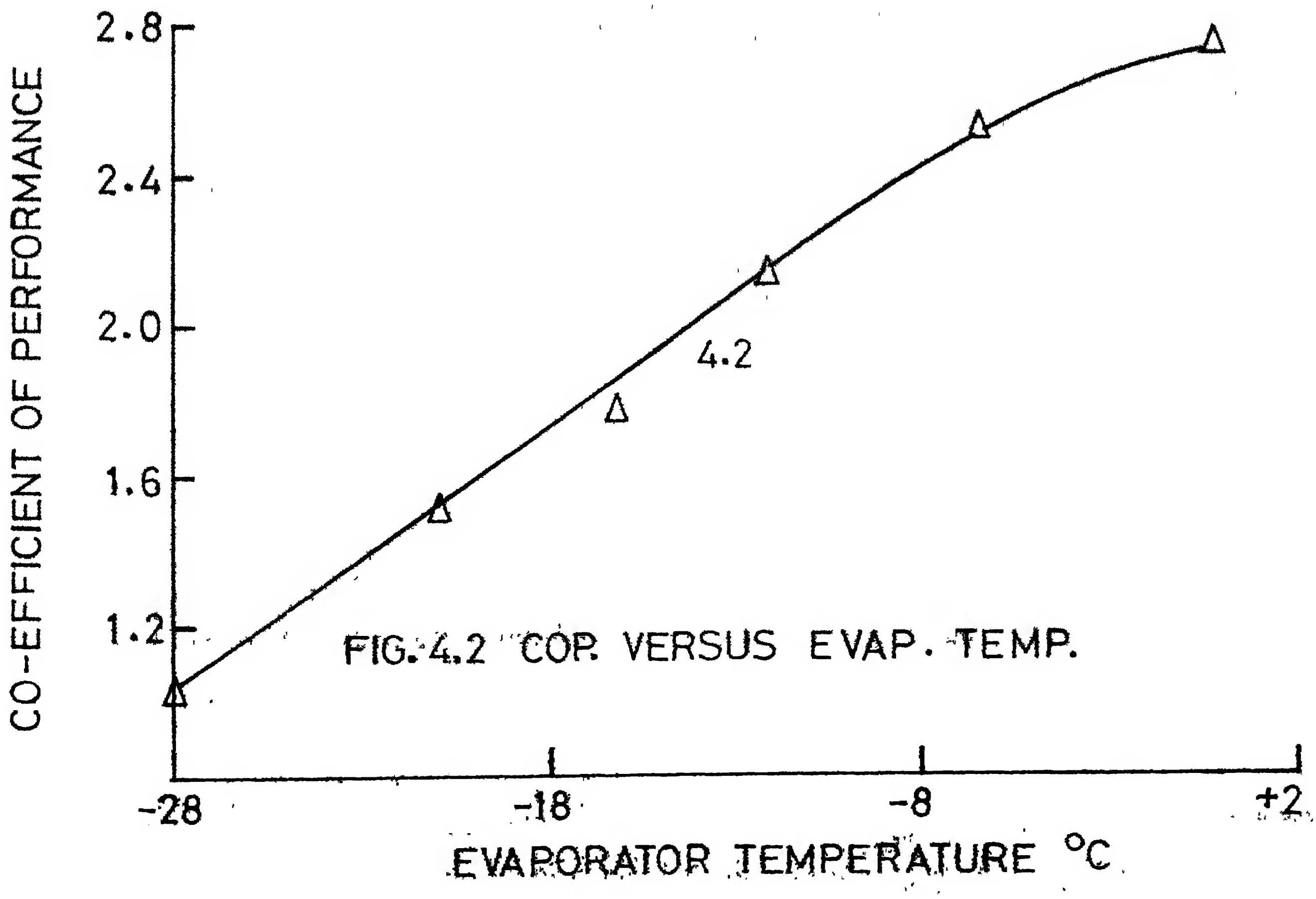


FIG. 4.2 COP. VERSUS EVAP. TEMP.

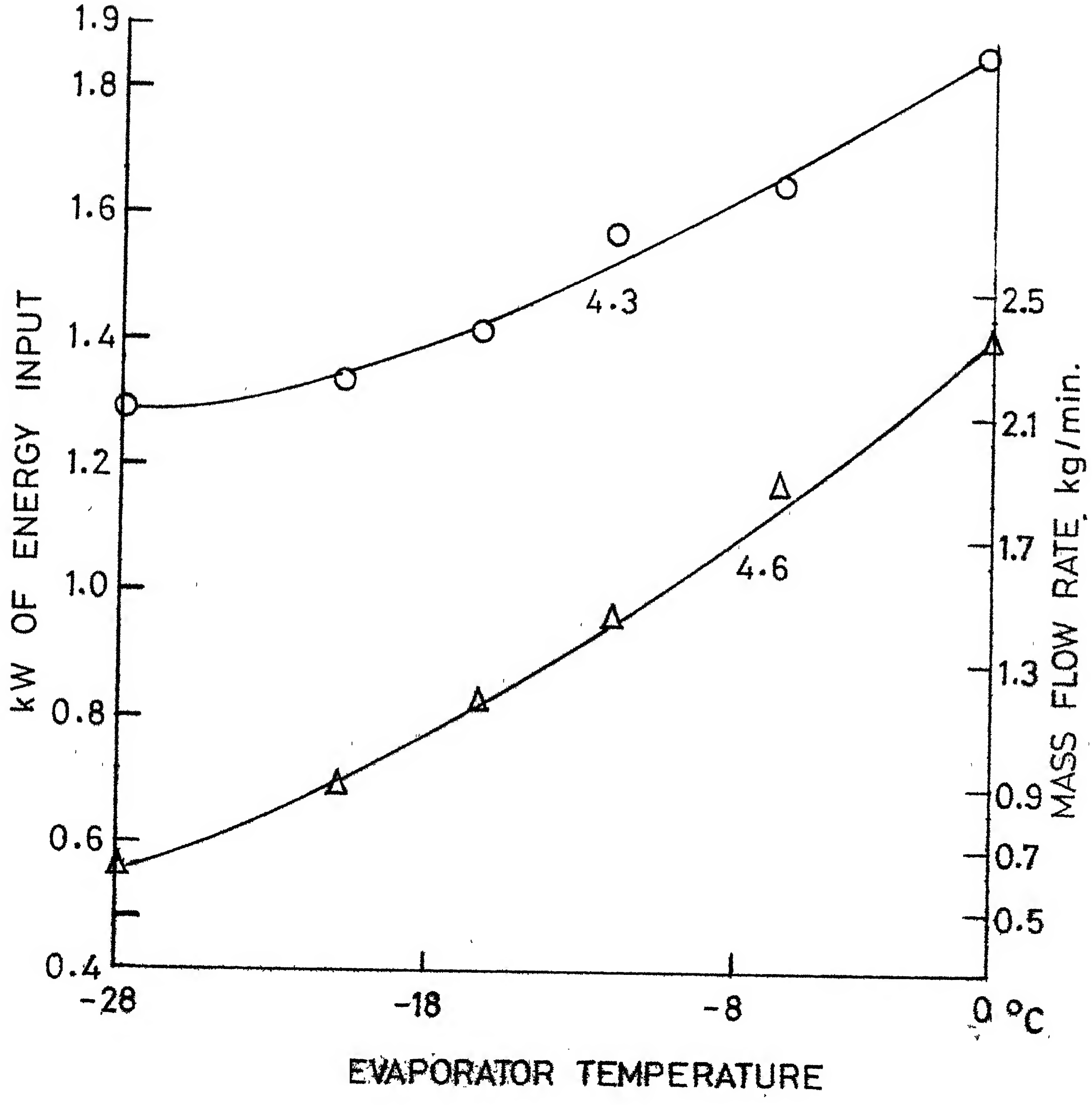
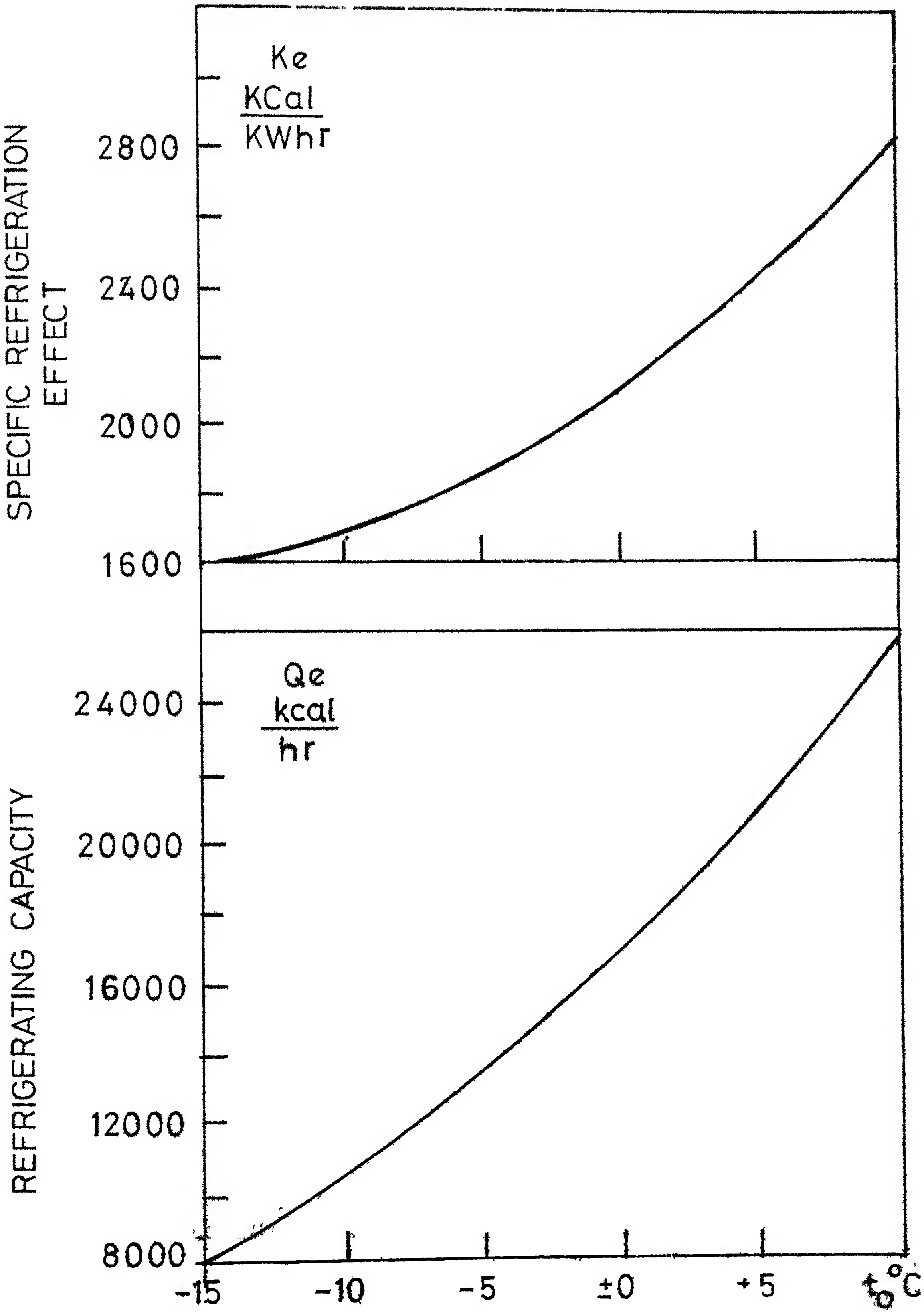


FIG. 4.6 MASS FLOW RATE VERSUS EVAP. TEMP.

FIG. 4.3 ENERGY INPUT VERSUS EVAP. TEMP.



t_C = 50 °C.C = 4.66 %

FIG.4.4 RESULTS OF CHAIKOVSKY ETC. [3]

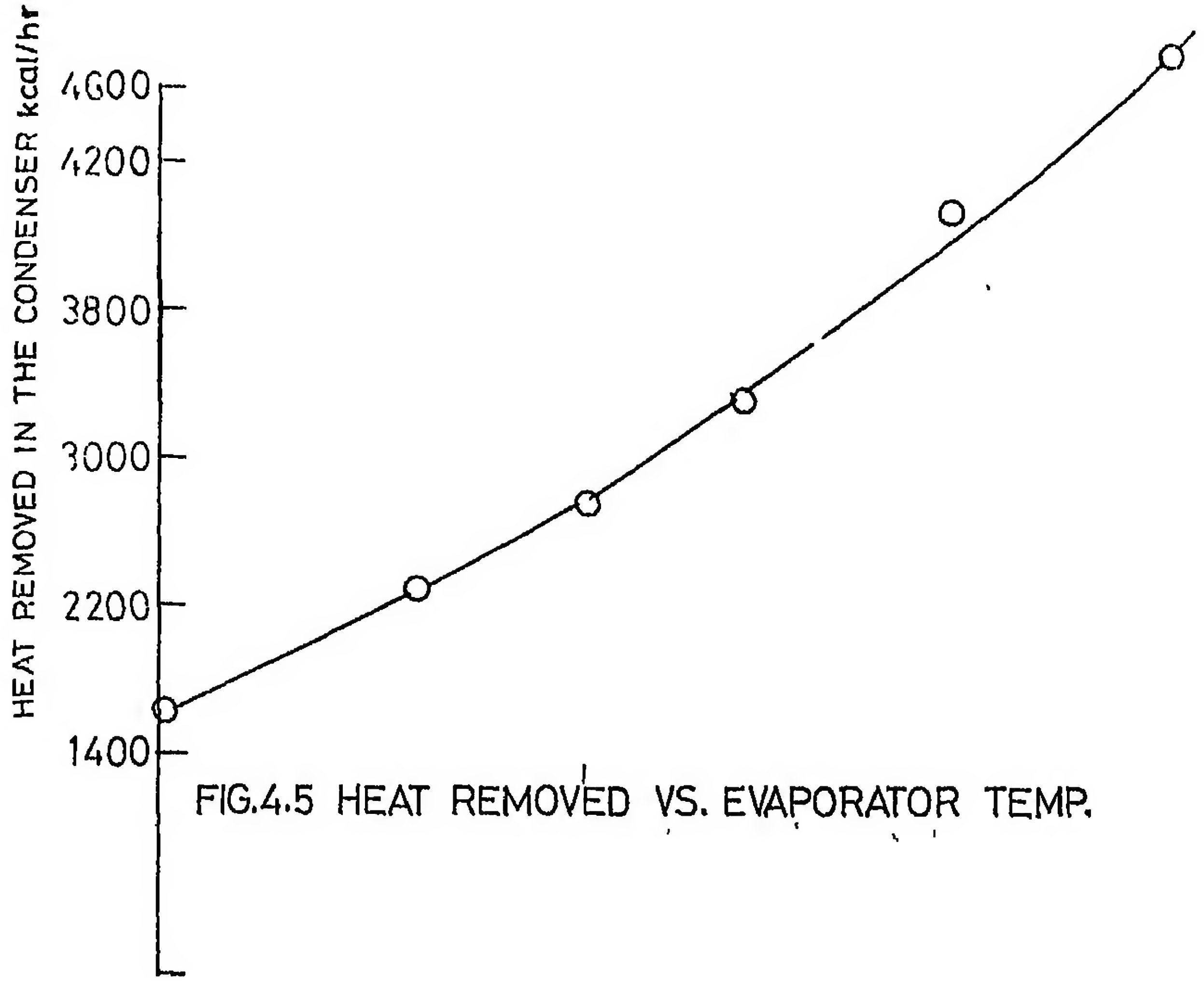


FIG.4.5 HEAT REMOVED VS. EVAPORATOR TEMP.

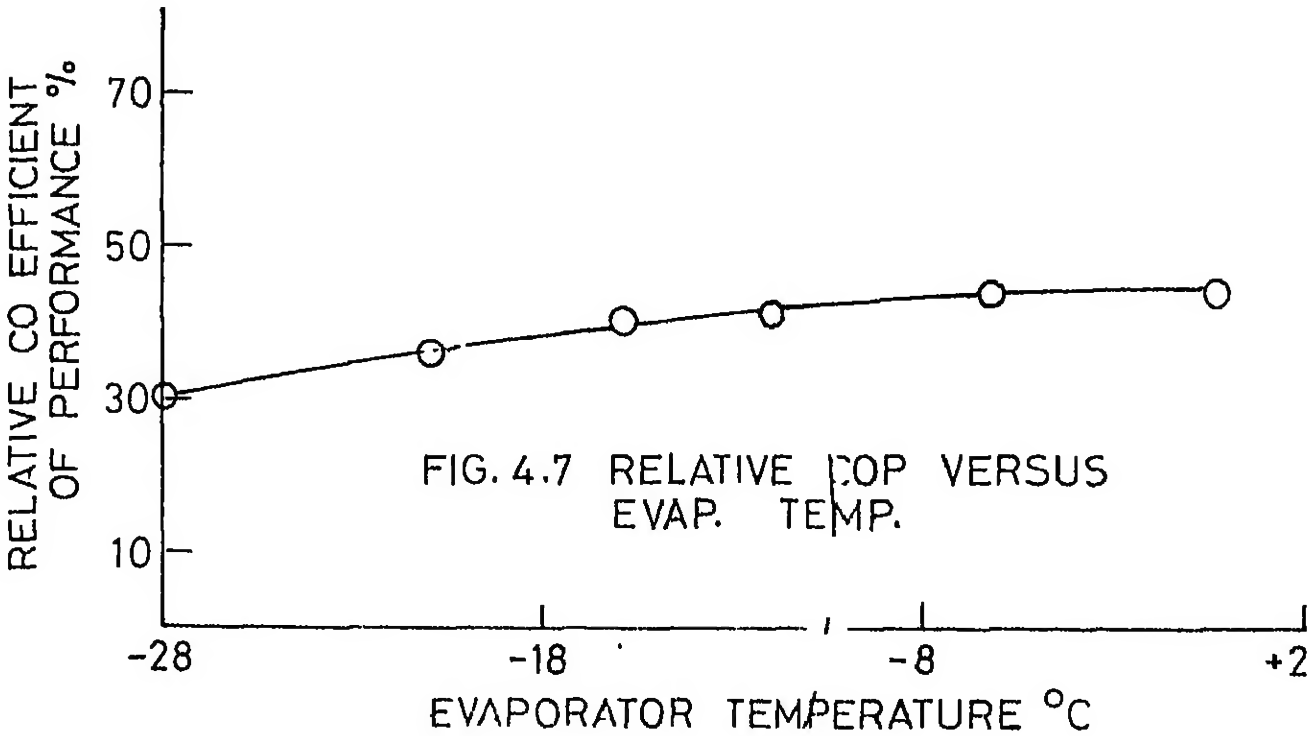


FIG. 4.7 RELATIVE COP VERSUS EVAP. TEMP.

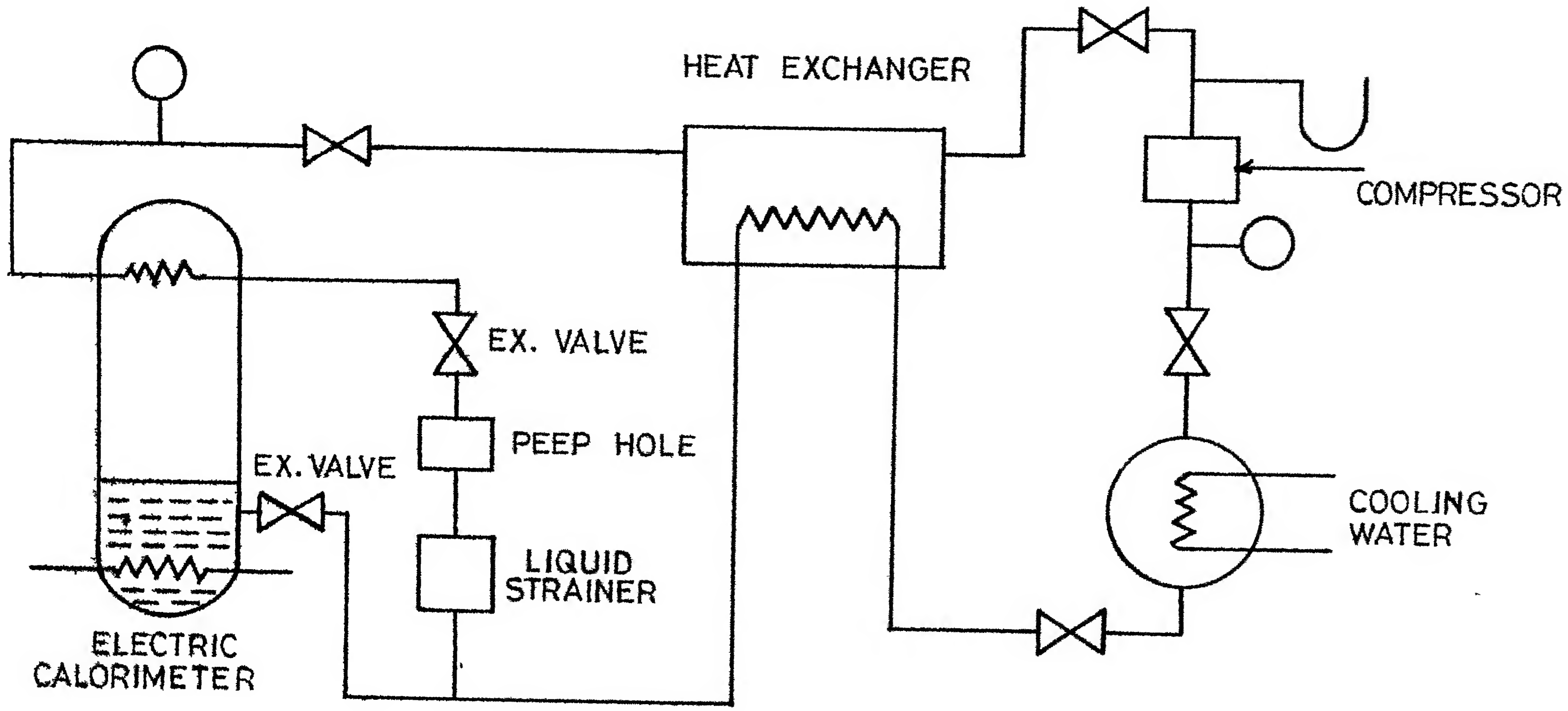


FIG. 4.8 EXPERIMENTAL SET-UP [3]

APPENDIX A-1

CALIBRATION OF INSTRUMENTS

A-1.1 Calibration of Rotameter

A 10.5 gpm rotameter is calibrated by passing tap water through a valve and collecting water in a bucket during a known period of time. The weight of the collected water is determined by using a spring balance (least count 12.5 gms).

The calibration curve is exhibited in Fig. A-1.1

A-1.2 Calibration of Liquid Refrigerant Flow Meter

Both the vessels FL1 and FL2 of the liquid refrigerant flow meter are calibrated by passing demineralized water. A measured volume of water is poured in the vessel and the height of the liquid level is measured in the graduated gauge glass of the flow meter. This process is continued untill the level reaches the upper limit of the vessel.

The calibration curve is shown in the Fig. A-1.2(a) for the FL1 and Fig. A-1.2(b) for FL2.

A-1.3 Calibration of Thermocouple

24 BSW gauge copper-constantum thermocouples are used. The calibration curve is shown in Fig. A-1.3.

A-1.4 Calibration of Pressure Gauges

The pressure gauge, to be calibrated is fitted to the dead-weight pressure gauge by using a suitable adopter. The air bubbles are removed from the oil tank very carefully after

running the piston back and forth several times and this process is continued till no bubble is seen. Similarly the air bubbles in the valves are removed.

Then gradually the weights are put on the disc and the pressure is raised by screwing the piston inside till the disc carrying the weight is lifted up. The readings of the pressure gauges are noted.

Special care is taken to see that the oil does not leak from any point. The readings are repeated and the calibration curves are shown in the Fig. A-1.4(a) and Fig. A-1.4(b).

A-1.5 Calibration of the Three Phase Energy Meter

The three phase energy meter (EG2) is calibrated under a load of power factor nearly unity. The voltmeter and an ammeter of the required range are put in the circuit. The loads are varied to take several sets of the readings and the calibration curve is exhibited in Fig. A-1.5.

A-1.6 Calibration of the Single Phase Energy Meter

The single phase energy meter (EG) is calibrated under a load of power factor nearly unity. A voltmeter and an mmeter are used in the circuit. The calibration curve is exhibited in Fig. A-1.6.

Calibration of the Calorimeter

The calorimeter is calibrated according to the specification laid down by Indian Standards Institution and the procedure is given as follows :

After insulating the calorimeter, it is filled with liquid refrigerant upto the operating level. The outlet and the inlet valves of the calorimeter are closed. The electrical energy is supplied to it. The input is so maintained that the evaporator temperature becomes about 14 °C above ambient. Steady state is supposed to be reached when there is only 0.4 °C variation in temperature in one hour. Then the power input devided by the temperature difference between the refrigerant and the ambient turns out to be 0.45 Kcal/min. This is the heat leakage factor F_1.

APPENDIX A-3

A sample of calculation is shown in the following.

The data are given in the table A-3.

The energy input to the calorimeter by loading device $= (4.54 \times 860) / 60.0 = 65.0$ kcal/min.

Heat leakage from the atmosphere

$$(0.45)\ (19.5 + 0.5) = 9 \text{ kcal/min.}$$

Total energy input to the calorimeter = 74 kcal/min.

The enthalpy (refrigerant coming out of the evaporator) $h_{g1} = 137.7$ kcal/kg and that of the subcooled liquid coming out of the condenser, h_{f1} is 106.1 kcal/kg.

Then the mass flow rate of the liquid

$$\dot{m}_f = \left[\phi_h + F_1\ (t_a - t_r)\right] / \ (h_{g1} - h_{f1}) = 74.0/31.6$$
$$= 2.34 \text{ kcal/min.}$$

The enthalpy of the saturated liquid at the compressor pressure : $h_{f2} = 107.4$ kcal/kg.

Then the refrigerating capacity, RF $= \dot{m}_f\ (h_{g1} - h_{f2})$ $= 71$ kcal/min and 4260 kcal/hr.

Power input to the motor, $P = 2.51$ kW

Motor losses : $\left[(2.51/3.5)^2\ (600) + 300\right] / 1000 = 0.612$ kW.

Net energy available at the motor shaft is $2.51 - 0.612 = 1.898$ kW.

Assuming a transmission efficiency of 95% the energy input to the compressor shaft is $(1.898)\ (0.95) = 1.8$ kW $= 25.7$ kcal/min.

Co-efficient of performance :

$$\text{COP} = 71 / 25.7 = 2.76$$

Flow meter reading is 80 cm rise in level in 2.05 minute.

Level rise per minute is 39 cm = 1950 cc/min.

The density of the refrigerant at 25.49 °C is 0.001385 kg/cc.

The mass flow rate is 2.7 kg/min.

Deviation = 0.36 kg/min = 14.3%.

$(COP)_s$, the co-efficient of performance of the simple vapour compression refrigeration cycle, assuming suction to be dry saturated is calculated from p-h chart of R-12.

$$\text{Refrigeration effect} = (h_{g2} - h_{f1}) = 137 - 107.4$$

$$= 29.6 \text{ kcal/kg.}$$

$$\text{Isentropic work} = (141.5 - 137) = 4.5$$

$$(COP)_s = 29.6/4.5 = 6.35$$

$(COP)_R$, the relative COP is given as the ratio,

$$2.76/6.35 = .435 \text{ or } 43.5\%.$$

$$\text{Specific refrigeration capacity} = (COP)\,(860) = (2.76)\,(860)$$

$$= 2370 \text{ kcal/kW, hr.}$$

APPENDIX A-4
MOTOR LOSSES DETERMINATION

A-4.1 OPEN CIRCUIT TEST, Fig. A-4.1(a)

Voltage 415 (rated)

First W_1 = 205 x 5 = $\pm$ 1025

Second W_2 = −145 x 5 = − 725

Total Power = 1025 − 725 = 300 W

A-4.2 BLOCKED ROTOR TEST

First W_1 = 320 x 2 = 640 W

Second W_2 = −20 x 2 = −40 W

Total Power = 600 W

Voltage 104

I_c = 7.5

Copper losses in both stator and rotor at full load : 900 W

The loss in watts at any load can be calculated by the equation :

$$L = (\text{fraction of full load})^2 \, (600) + 300$$

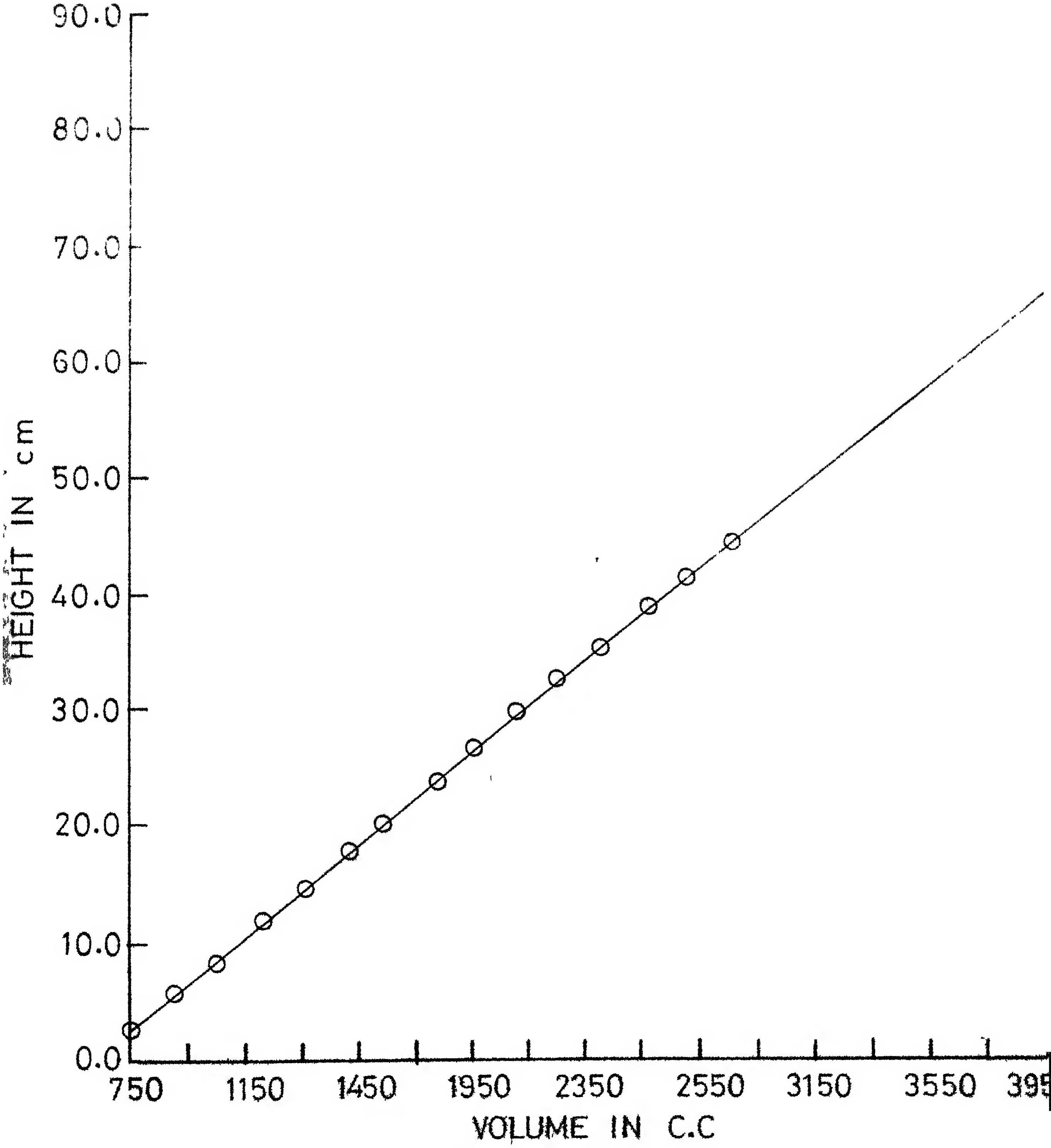
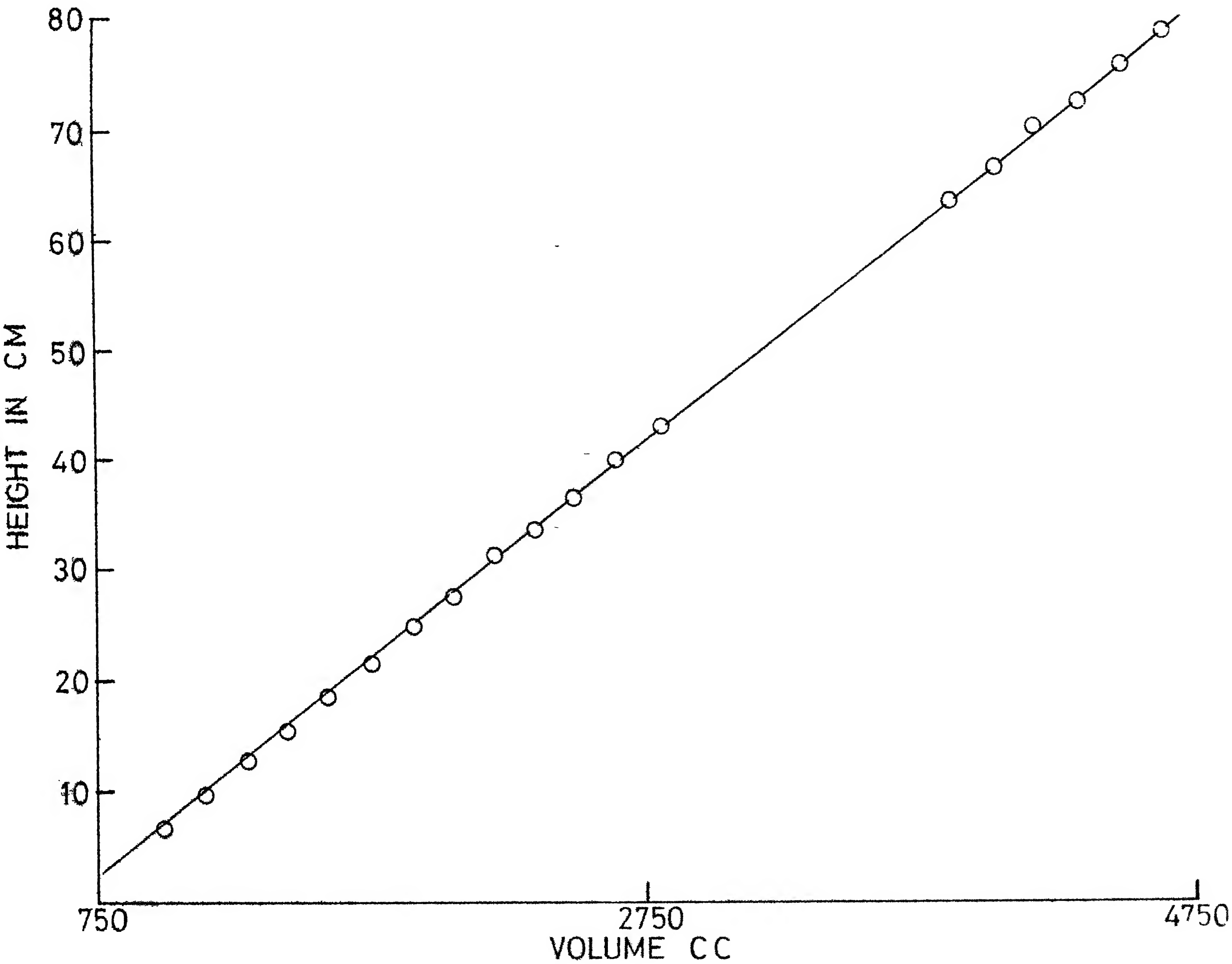


FIG. A-1.2 (a) CALIBRATION OF FL1 OF FLOW METER


80
70
60
50
40
30
20
10
HEIGHT IN CM
750
2750
4750
VOLUME CC

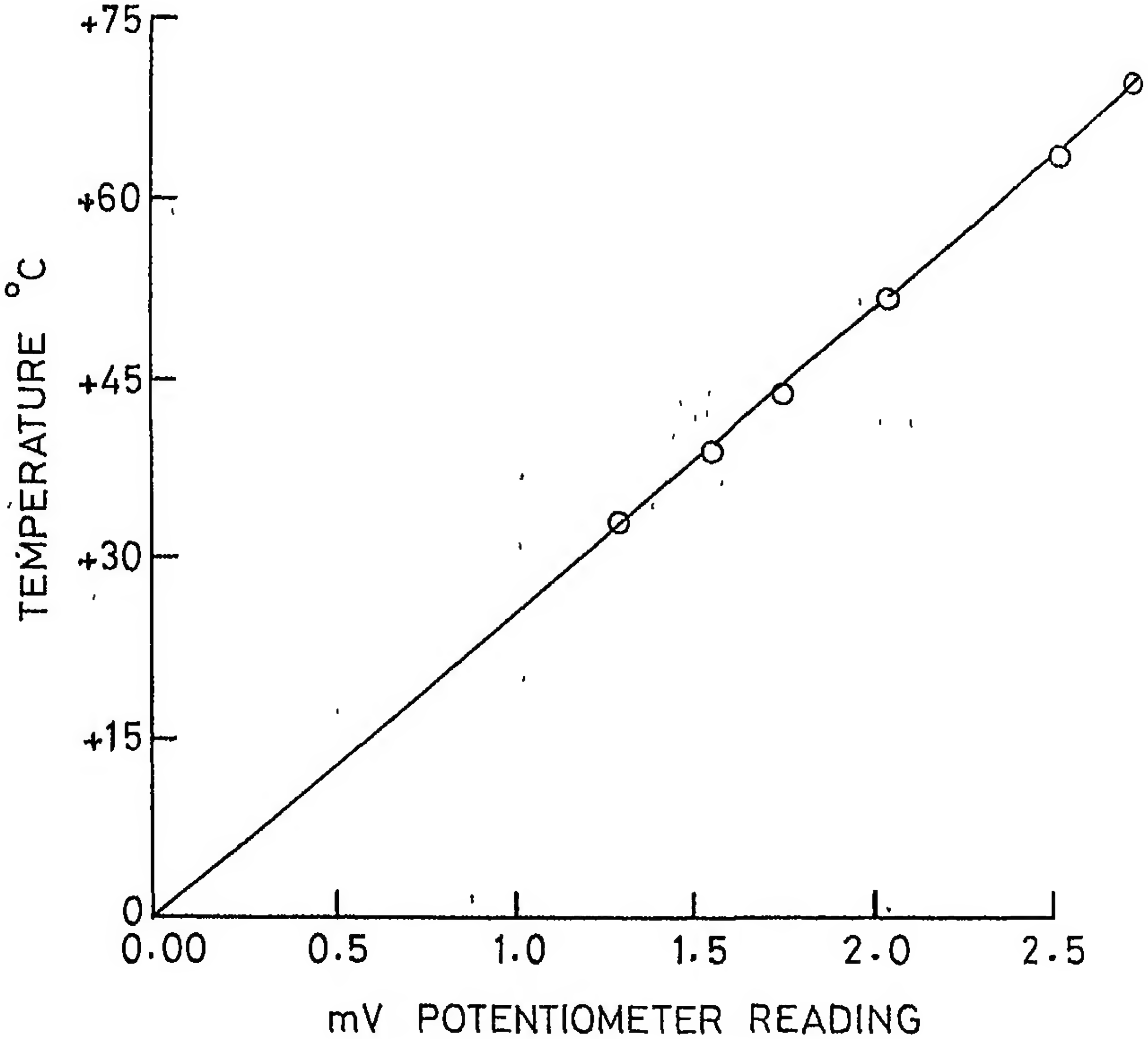


FIG. A-1.3 THERMOCOUPLE CALIBRATION

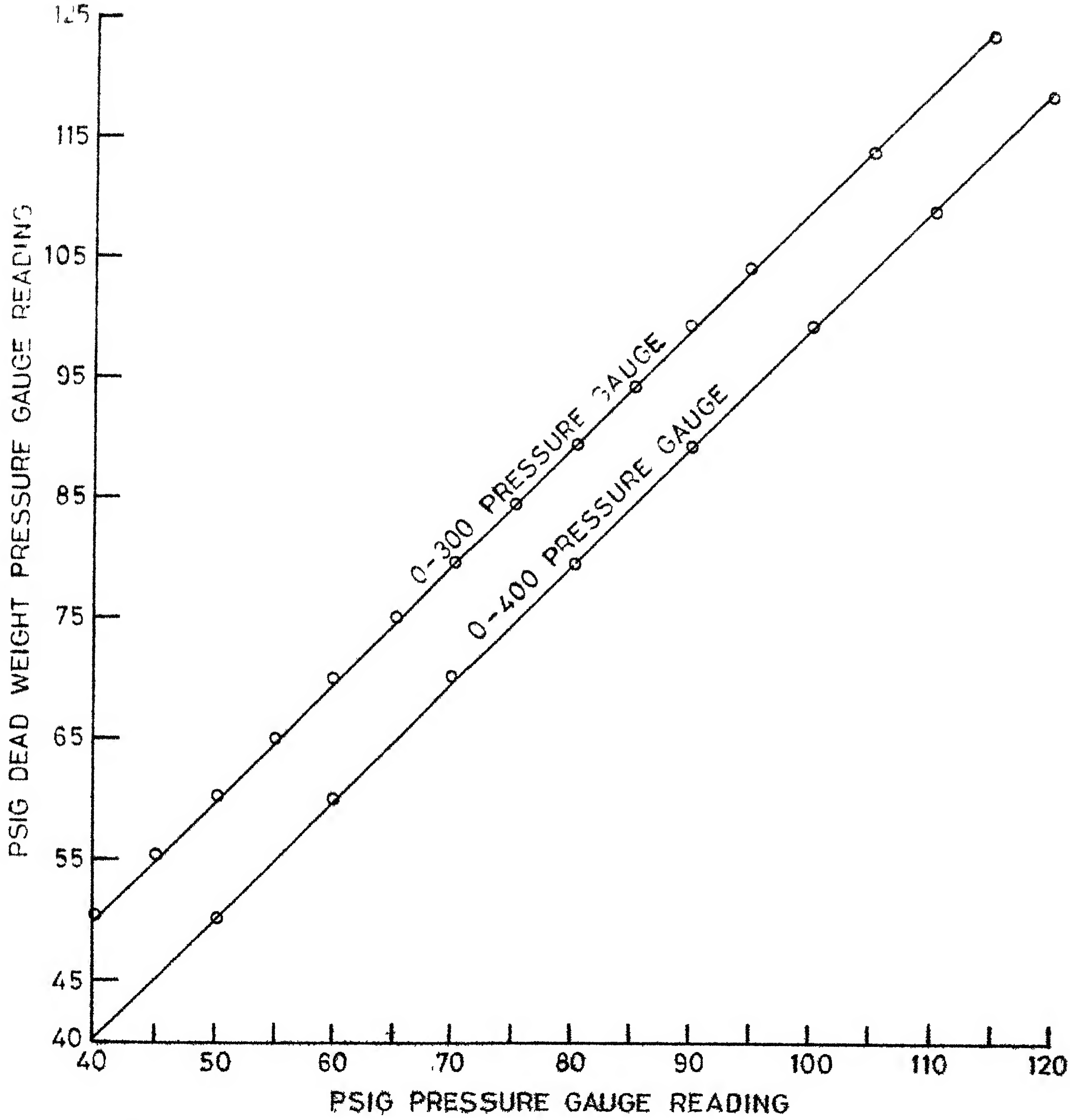


FIG. A-1.4(b) PRESSURE GAUGE CALIBRATION CURVE

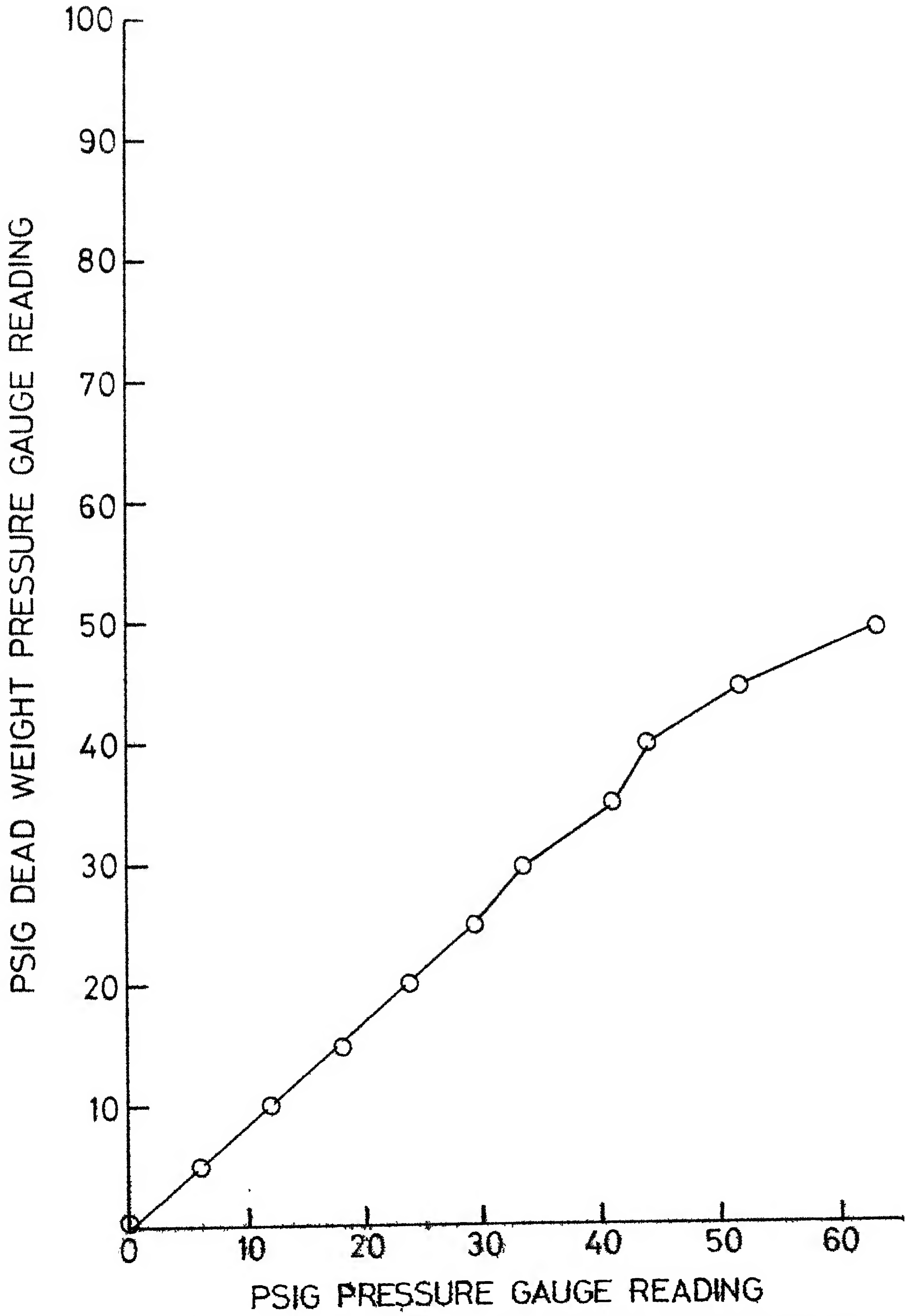


FIG. A-1.4(b) PRESSURE GAUGE CALIBRATION CURVE

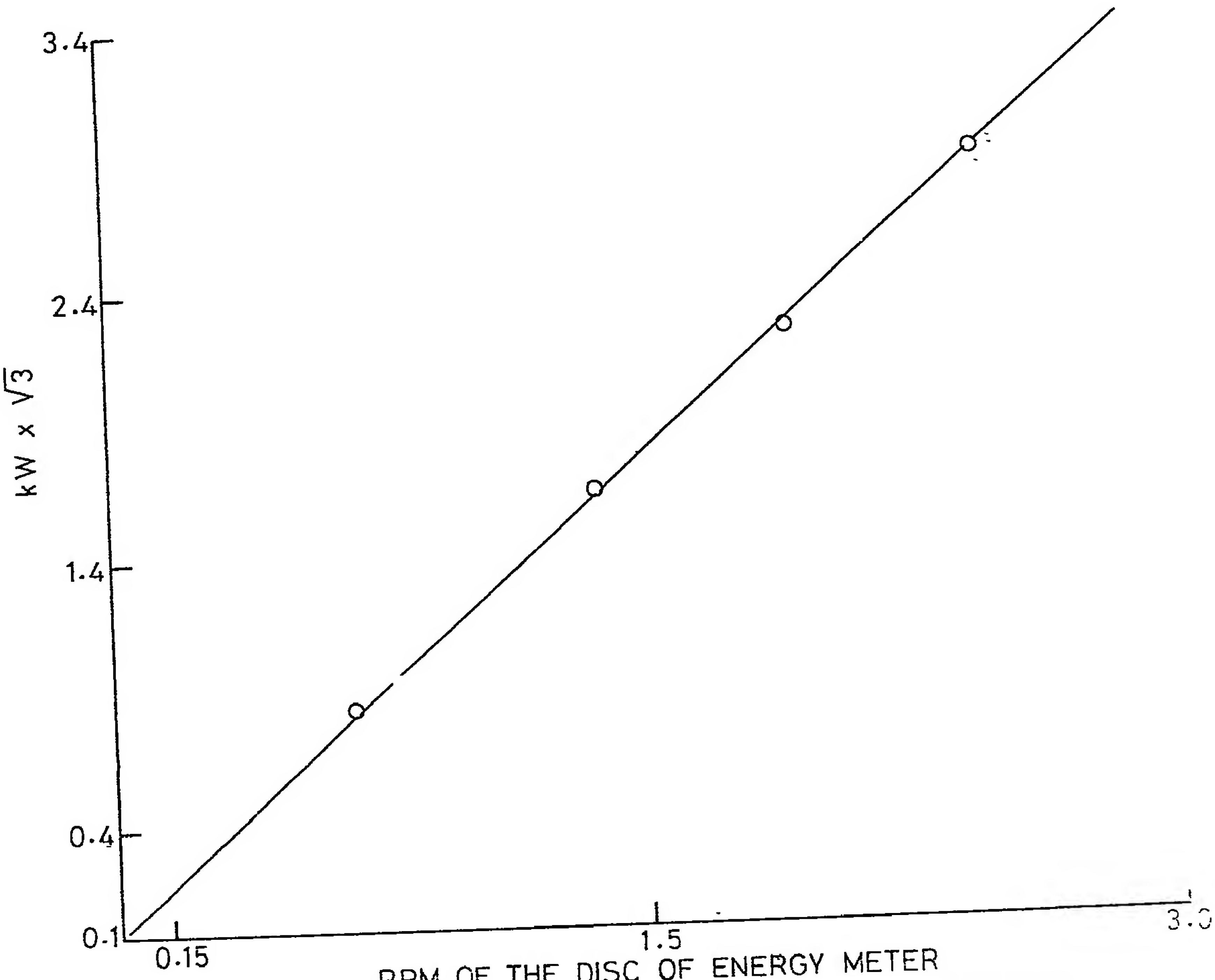


FIG. A-1.5 THREE PHASE ENERGY METER CALIBRATION

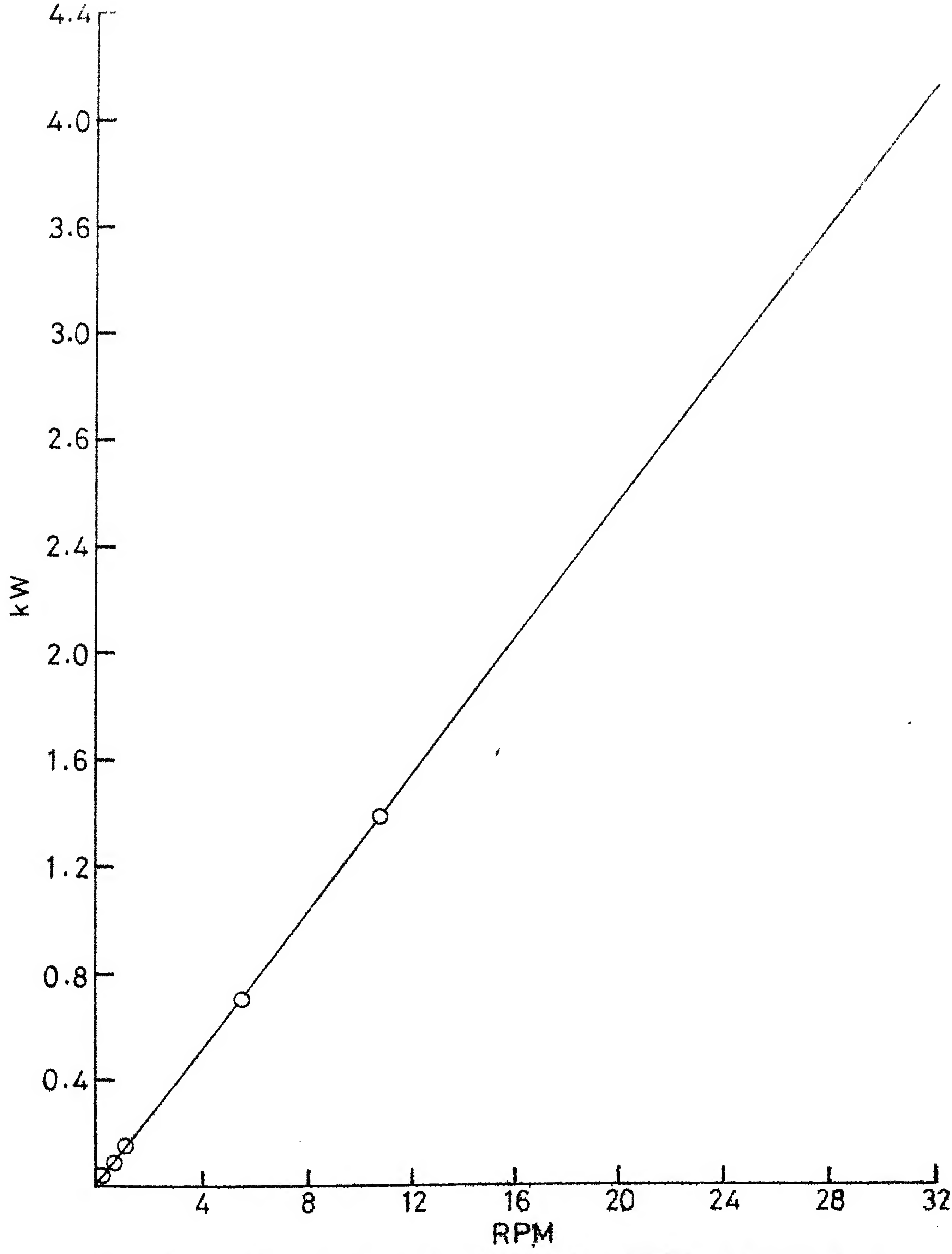


FIG. A-1.6 SINGLE PHASE ENERGY METER CALIBRATION CURVE

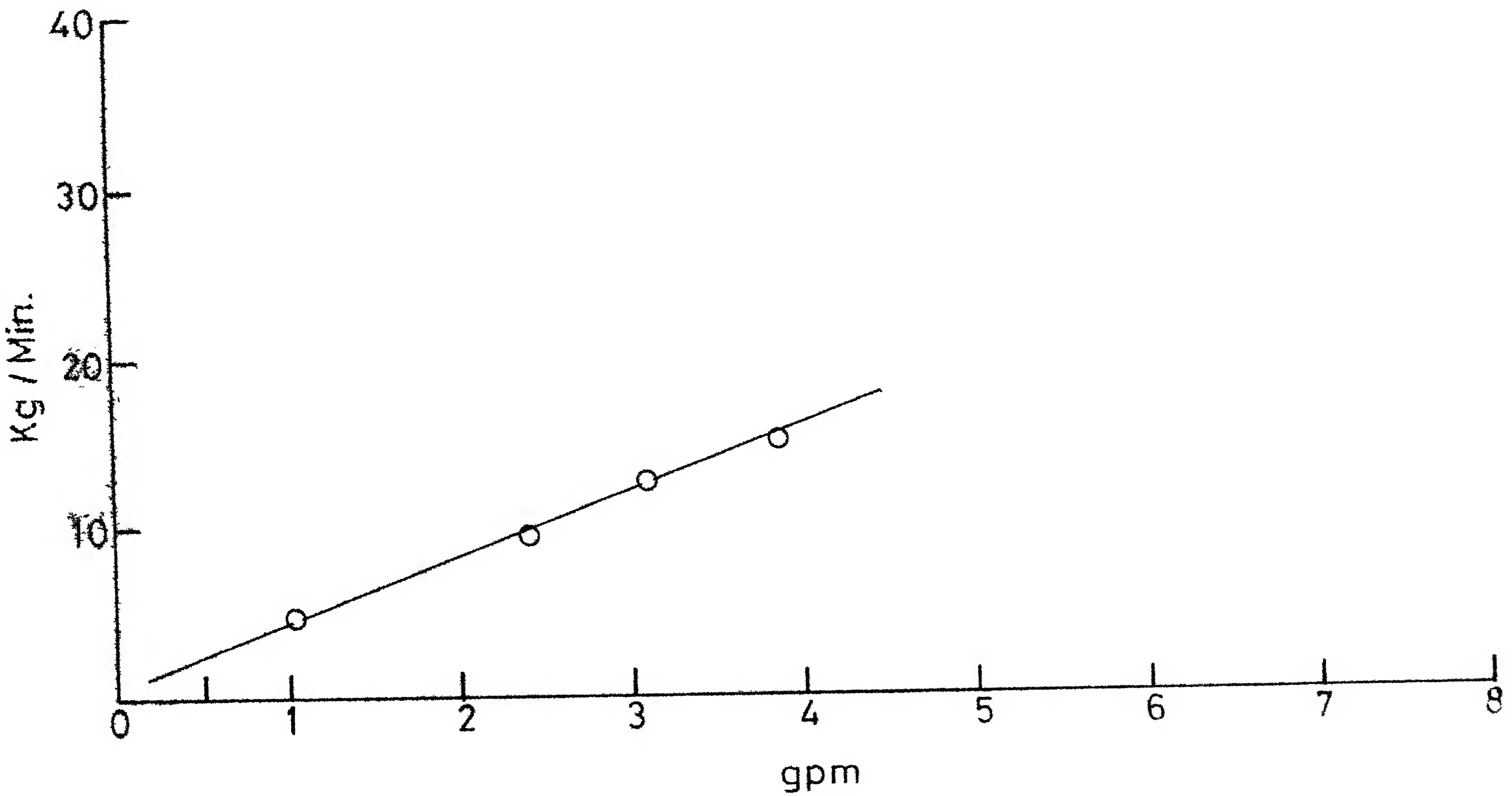


FIG. A-1.7 ROTAMETER CALIBRATION CURVE

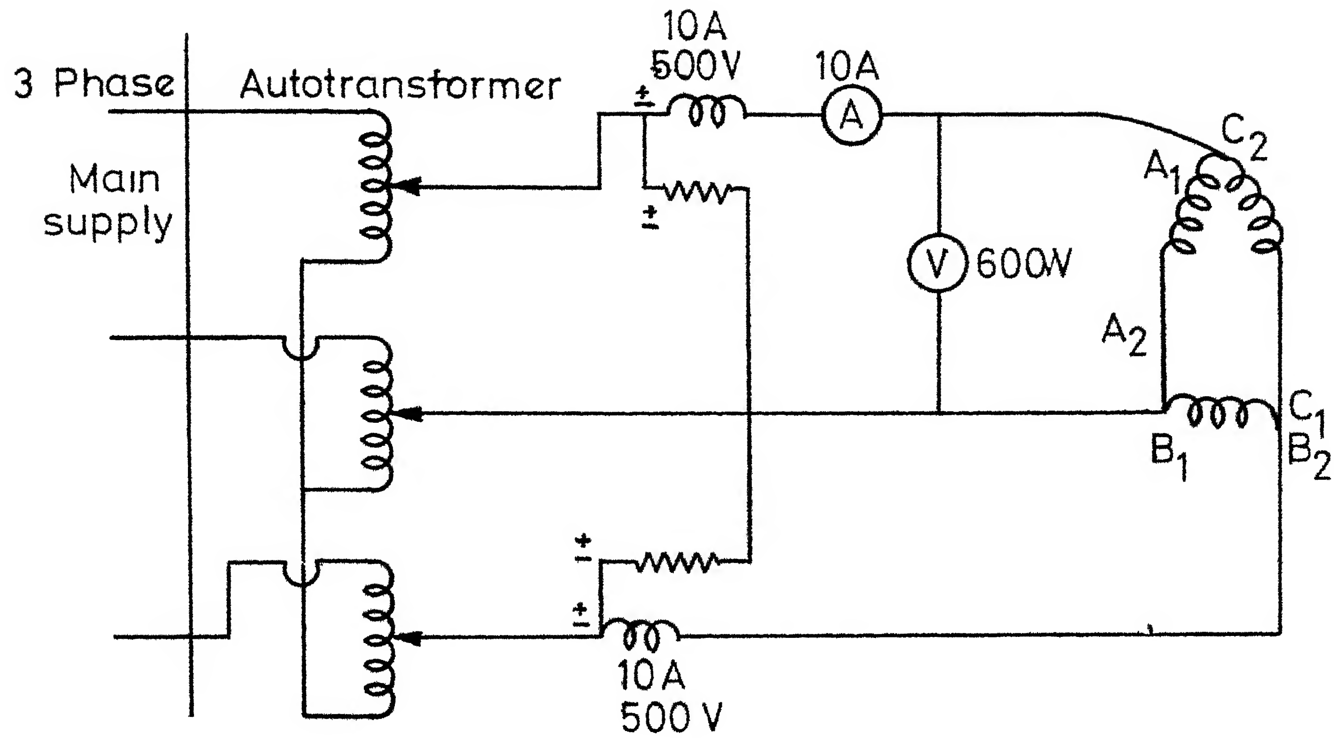


FIG. A-1.4 ELECTRICAL CIRCUIT DIAGRAM MOTOR LOSSES DETERMINATION

TABLE A-3

S.No.	Description	Reading Observed	Calibrated Values
1	P_1	100 PSIG	8.06 kg/cm^2
2	P_2	100 PSIG	8.06 kg/cm^2
3	P_3	90 PSIG	8.06 kg/cm^2
4	P_4	90 PSIG	8.06 kg/cm^2
5	P_5	35 PSIG	3.08 kg/cm^2
6	P_6	35 PSIG	3.08 kg/cm^2
7	TC_1	1.77 mV	64.54 °C
8	TC_2	.914 mV	42.8 °C
9	TC_3	.238 mV	25.5 °C
10	TC_4	.237 mV	25.49 °C.
11	TC_5	-.782 mV	-.49 °C
12	TC_6	-.565 mV	+4.95 °C
13	kW_2	8 Min 7.8 Sec. (10 REV.)	2.51 kW
14	kW_1	17.3 Sec. (10 REV.)	4.54 kW

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